The George W. Woodruff
School of Mechanical Engineering

AUTOMATED REGOLITH MOVEMENT SYSTEM

December 1987

Georgia Institute of Technology
Atlanta, Georgia 30332

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James Brazell
Professor
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Dear Mr. Brazell:

Accompanying this letter is the Mechanical Design Engineering Report of the Automated Regolith Movement System. The report examines the systems involved in the collection and throwing mechanisms of the ARMS vehicle. We would like to thank you, Gary McMurray, and Bryce Maclaren for your assistance and guidance during all phases of this design.

Sincerely,

Andrew Kates, Group Leader

Paul Crawford

Michael A. DiMarco

Ben Hagler

Valerie L. McWhorter

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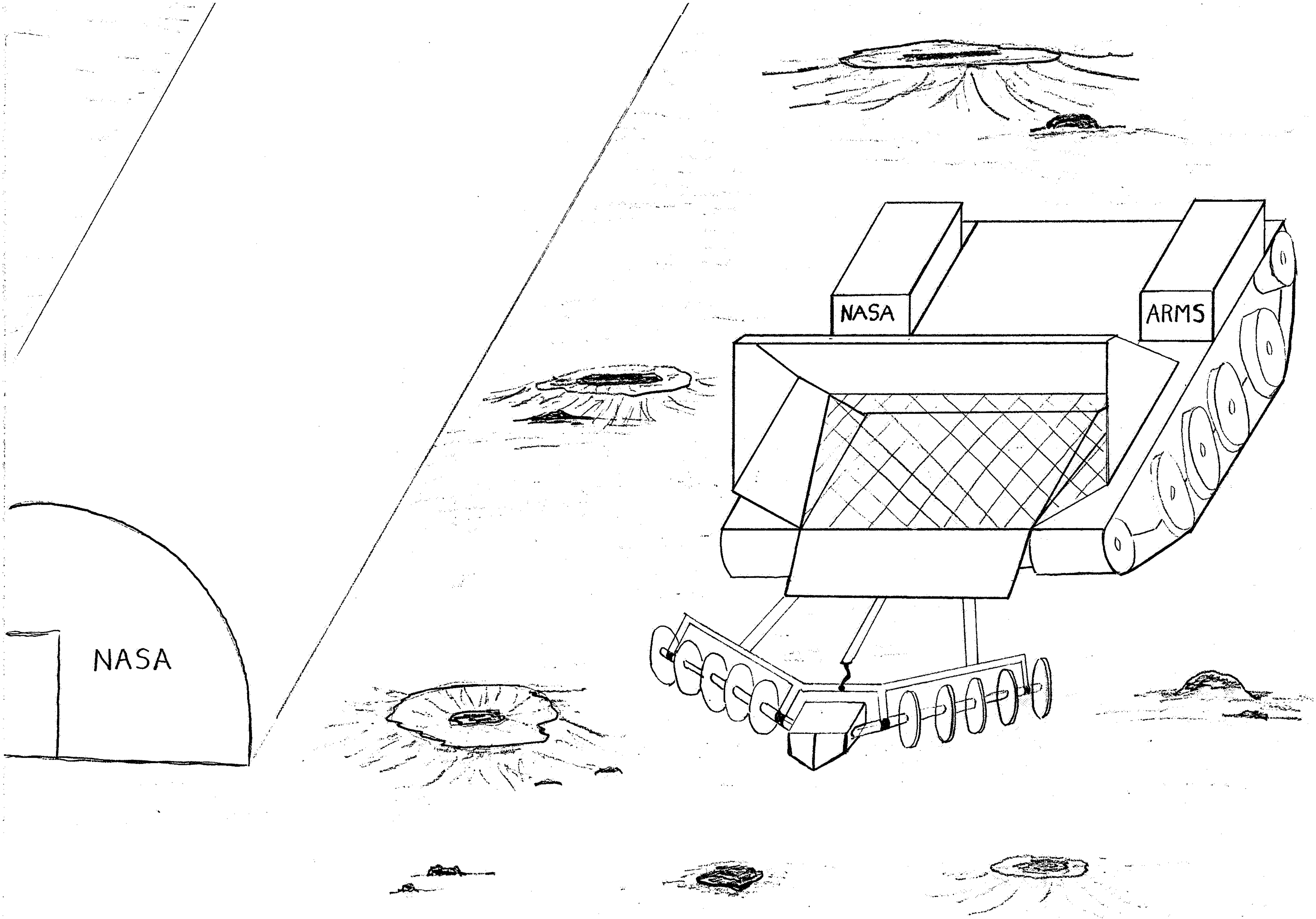
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AUTOMATED REGOLITH MOVEMENT SYSTEM

(SIDE VIEW)

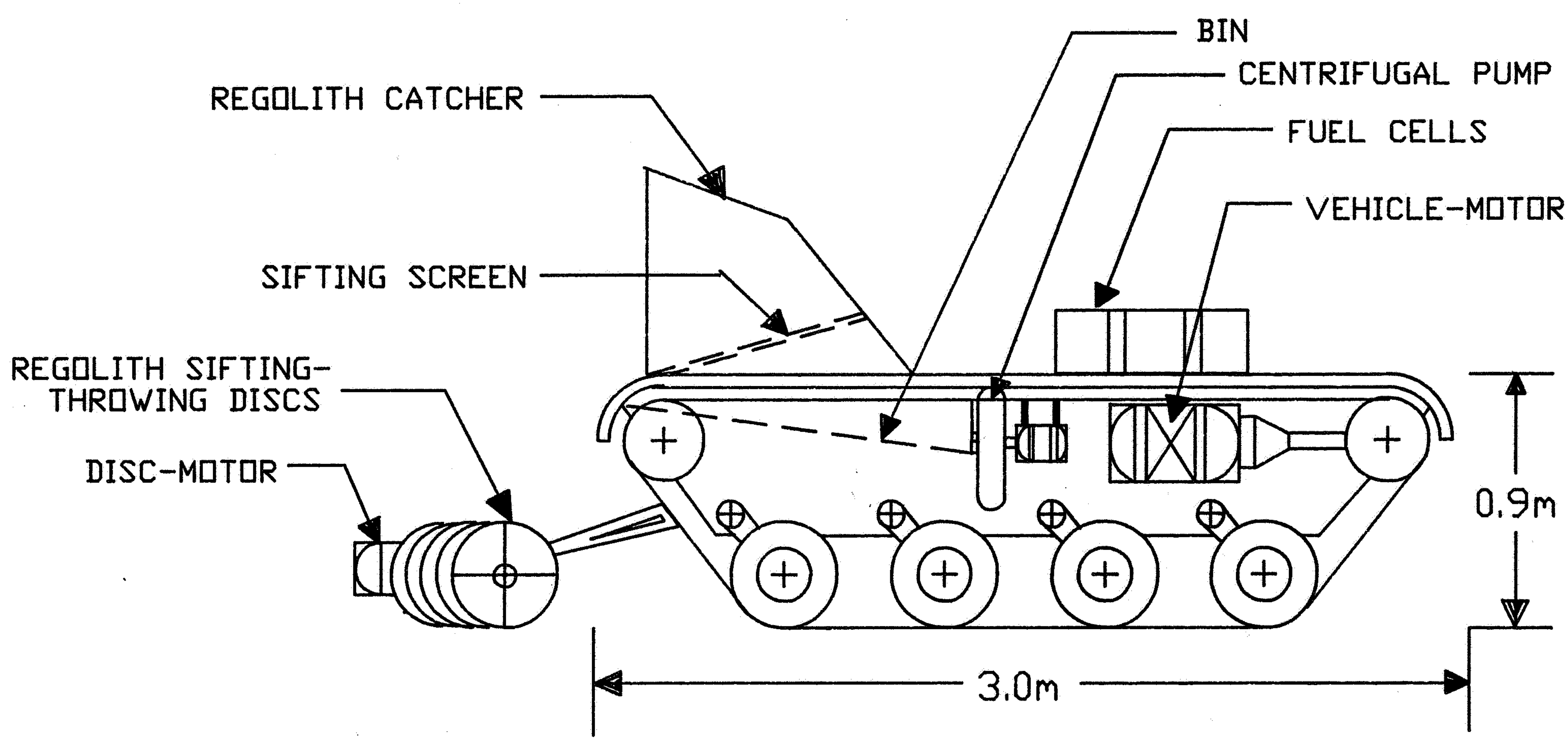

Figure 1

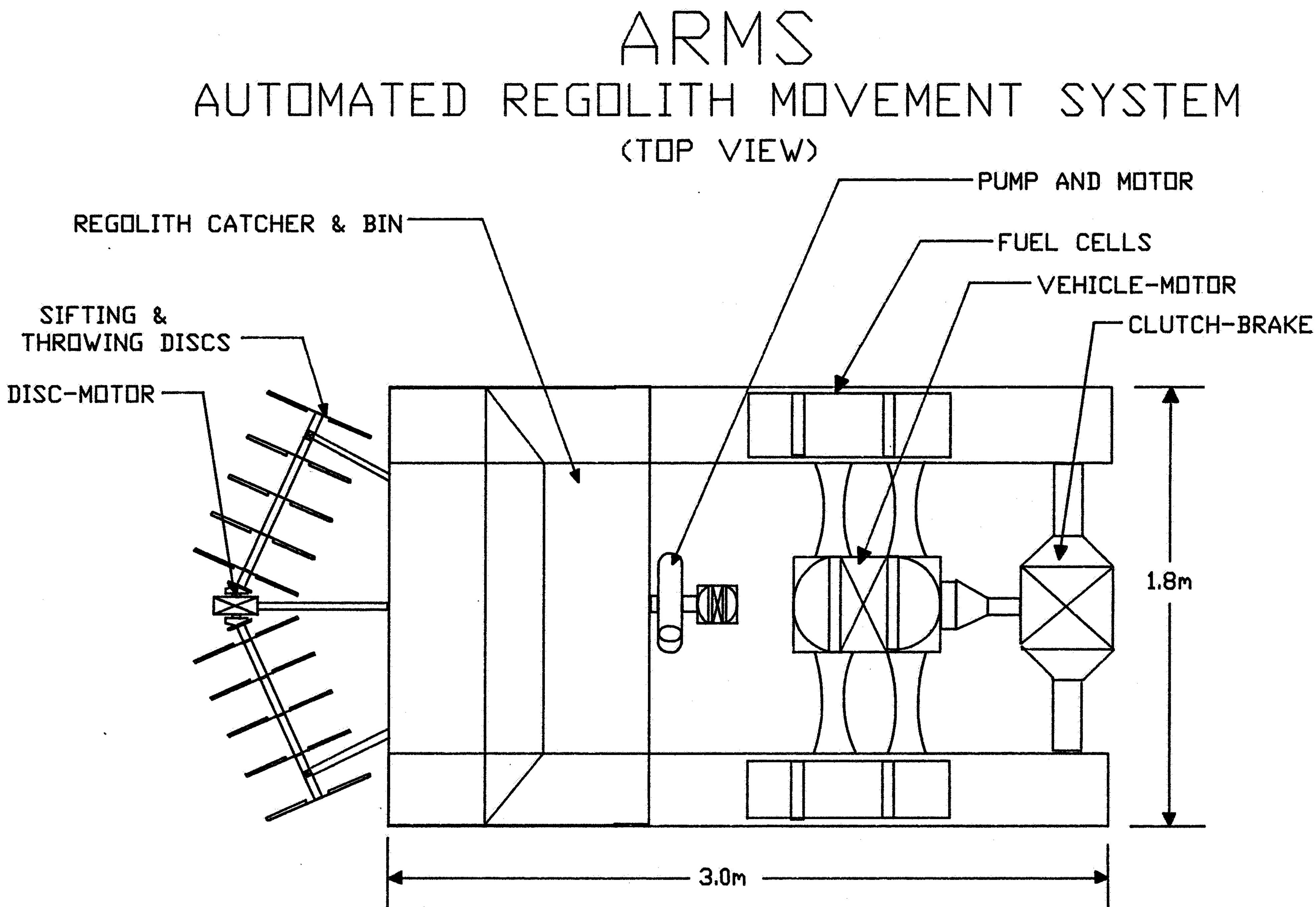
ARMS
AUTOMATED REGOLITH MOVEMENT SYSTEM
(TOP VIEW)
PUMP AND MOTOR
REGOLITH CATCHER & BIN
FUEL CELLS
VEHICLE-MOTOR
CLUTCH-BRAKE
SIFTING &
THROWING DISCS
DISC-MOTOR
1.8m
3.0m

Figure 1

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AUTOMATED REGOLITH MOVEMENT SYSTEM ABSTRACT

This project consists of the design of an autonomous system to bury lunar modules with regolith. This system is broken down into a collection system and a throwing system. Both functions are incorporated into one device that will roam around a lunar module.

The collection will be done by a series of vertical disks mounted on the front of the vehicle that will throw the soil into a bin. The soil will then be thrown toward the module by a solid handling pump mounted on the back of the bin. Both operations will take place will the device follows a symmetrical pattern around the module.

To adequately bury a lunar module, 40,000 ft^3 of regolith is required. The device is designed to bury one module in twelve Earth days.

PROBLEM STATEMENT

The intense radiation and temperature extremes of the lunar surface can be damaging to modules placed on the moon. One way to protect these modules is to bury them with regolith. In order to accomplish this, a devise is required that can move regolith from the area surrounding the module to on top and around the module.

The device must be fully automated to allow for the unmanned deployment of modules. In addition, the device must be able to withstand the harsh conditions on the moon. During the lunar "day", the temperature can vary from roughly 400 K on the sunny side to 105 K on the shaded side of the device. The mechanism must be able to operate in a vacuum atmosphere. Therefore no liquid lubricants can be used. The device must also be impermeable to dust since the operation of the machine will create large amounts of dust.

To adequately cover the module, a layer of regolith 2 meters thick is required. In order not to damage the module, this layer must consist of only soil and small rocks. Roughly 30,000 ft^3 of soil is required for each module. This will require that the device can move the soil from 260 ft on either side of the module to on top of the module. The device should be able to cover one module in one lunar "day" which is roughly 14 earth days.

To minimize transportation costs, this device should be as small and lightweight as possible. Material costs and building costs are minimal in comparison to transportation costs. Extremely high reliability is also required since this device is to be unmanned.

INTRODUCTION

NASA has plans to build a base on the moon in the next decade. Such a base will be made up of cylindrical modules. These module must be protected from the harsh lunar environment. Since the moon has no atmosphere, the surface is subject to both extremely high temperatures (270 F) in direct sunlight and extremely cold temperatures (-270 F) in the shade as well as intense radiation. Additionally, micro-meteorite storms have a sandblasting effect. One way to protect the modules is to cover them with regolith.

This project consists of the design of a device that can cover the modules with regolith thus protecting them. Numerous studies were done to add in the design of the device. These studies included work on lunar trajectory analysis, impact mechanics, lunar terrain, and regolith properties. Experiments were conducted on design ideas and impact forces.

To adequately cover a module, regolith must be gathered from the surrounding lunar surface and thrown on top of the module. Thus the device must both collect regolith and throw it. Additionally, only fine regolith and small rocks can be thrown so the device must also sort the collected material. The system should be autonomous to allow for unmanned deployment of the modules.

The proposed design is a treaded vehicle with both a collection system and a throwing system. It is designed to cover a module in 12 earth days. The total weight of the device is roughly 1,500 pounds. The mechanism measures 10 feet long by 6 feet wide.

1.0 VEHICLE

1.1 Terrain Summary:

The lunar terrain is strewn with craters and rocks of various sizes, posing a substantial problem to any vehicle design. Choosing a lunar base site that is flat and clear as possible will simplify all designs of base vehicles greatly. Possible base sites center on Apollo landings, since we already know the general geographic and geographical information there. The general information on these sites was taken from those Apollo mission reports, specifically 11, 12 and 15.

The optimum mission site for our vehicle will be most likely in a sea or "mare." Here the regolith, or lunar soil, is the oldest and deepest. Median regolith thickness ranges from 1 to 10 meters with 90% of the particles being less than 1 mm thickness. Crater and boulder sizes ranged from the very small, on the order of centimeters, to as large as 5 to 10 meters. Therefore we must design a vehicle capable of traversing these obstacles while maintaining the steadiest ride possible for optimum operation of the forward tilling device.

1.2 Vehicle Concept:

The overall design of our vehicle evolved from a wide variety of concepts. The major conflict was between wheels and tracks, or a track-layer vehicle. The lunar rover used metal mesh wheels with chevrons for traction. The slip experienced with this setup was in the neighborhood of 20%. Other design ideas involved using wheels including a rope and pulley-guided vehicle. The control scheme was simple but the system lacked flexibility in movement and was not practical. The design needed to be able to move on to the next job when finished with its first. Therefore it must be able to traverse the rugged lunar terrain, easily and relatively quickly. Legged walkers, such as the one done as a previous project, was not practical because of it's lack of smooth motion. Placing some kind of tiller and pump system under a walker would be difficult, seeing as the tiller would need to be in the soil continuously. A track-laying vehicle solved the problems of traction, mobility and simplicity better than all the other configurations.

1.3 Vehicle Suspension:

The suspension for our tracked vehicle needed to be simple, lightweight, strong and reliable. To maintain high reliability the simplest system was chosen. This would be the torsion bar system shown in Figure 2 . The torsion bar is lightweight and relatively inexpensive. The wheel load is made to apply a torque to a circular section aluminum bar whose twist in response provides the required compliance. Torsion bars, when made of proper materials, provide excellent strength and durability compared to the other systems, while being more reliable. Other systems included a bogey, coil springs, a three point system and the hydrogas shock absorber. The hydrogas system is not only bulky and heavy but also needs its own air supply which adds to the vehicle weight. This system requires too much maintenance, which will not be possible in our mission profile. Because the stress that can be employed in coil springs is reduced 25% of that for the torsion bar, it will be considerably heavier for a given resilience. Bogey systems work well at low speeds, but are considerably heavier than torsions bars while working only just as well.

A three-point suspension was also a possibility that showed simplicity and good performance. Problems came up in the actual design though, since it needs to be connected from the front (center) to the back of the vehicle. Our bin and pump command the space in the middle of the vehicle. Any linkage combination would need to be below the bin or through it. Below the bin would lower the hull clearance considerably and would complicate the overall design. Trying to go through the bin would not be practical, and the linkage design could never be as reliable as the simple torsion bars. Our forward disk assembly should help reduce the risk of running over large blocks.

1.4 Drive Train:

The components for the drive train were chosen to be simple and compact. The motor requirements for level ground travel were calculated to be a modest 2.8 hp. On an incline of 30 degrees the maximum motor power required was 3.76 hp (see Appendix 1). The motor type to be used will be an electrically powered torque motor of similar type to that used for the impeller. From the motor output the power will be transmitted to the centrifugal clutch. Figure 3 shows a friction clutch. A centrifugal clutch is simply an automated friction clutch using springs to restrain centrifugal motion. Such a system is reliable and transmits nearly 100% of the input power. The next component is the epicyclic gear-changer. This type is compact and simple maintaining high reliability. The epicyclic shown in Figure 4 features a soft gear shift, good for automatic control. This arrangement is also able to go through hot shifts, or the engaging of one gear ratio before before the previous one is completed. This transfers some, but not all, engine power throughout the duration of the shifts, saving power. To provide ideal operating conditions the gear systems will be hermetically sealed. This will provide a viable atmosphere for proper lubrication conditions.

1.5 Dimensions:

The dimensions of the vehicle play an important, if not an obvious, part in its performance characteristics. For tracked vehicles the length to width ratio L/C needs to be be kept within a certain range for acceptable steering. The range, $1.5 < L/C < 1.8$, helps to determine one parameter when the other is set. In our case a width of at least 6 feet was determined, which yields a minimum length of 9 feet. The final length was determined to be 10 feet, after iterations to determine an acceptable hull clearance (see Appendix 1). This long length also assures a stable ride over small craters of a few feet or less in width. The dimensions of the torsion bar suspension were chosen on the basis of materials and performance needs. A bar diameter of 2 inches was determined from the material strength for aluminum. Wheel size and design were chosen mainly on the basis of weight. Road wheel diameters of 18 inches and thickness of a quarter of an inch were determined on the basis of hull clearance and weight. The larger the wheel diameter, the higher the hull clearance.

1.6 Vehicle Performance:

Performance characteristics had to be evaluated analytically due to a lack of any testing. The important aspects of lunar travel are traction and obstacle avoidance. The accepted median value for a regolith friction coefficient is 0.649. Using this value it was found that to climb over any obstacle that obstacle had to have a friction coefficient of at least 0.3245. (see Appendix 1). Otherwise the vehicle will simply slip on it. Depending on the amount of slip, a traction force can be calculated (see Appendix 1) and a plot generated (see Fig. 5). A maximum slope angle was not determined from this traction data. Such numbers can only be determined by tests on active use. Past NASA missions have suggested angles of 25 degrees - 30 degrees possible with the lunar rover. The maximum height of an obstacle our vehicle can overcome is derived in the Appendix 1. Using numerous iterations an acceptable set of dimensions was determined and the maximum step height was found to be about 4.24 feet, or about 1.3 meters. To help prevent slip on these obstacles a minimum vehicle angle (beta) is kept low. At angles of beta greater than 45 degrees the slip on the obstacle will overcome that at the ground, obviously impeding the vehicle's travel. POwer requirements increased, as can be expected, with ground slope angle. The maximum power at the maximum slope was found to be around 1.7 hp. This was worked backward through the drive train to come up with the maximum motor power needed, previously given as 3.76 hp (see Figure 6).

1.7 Materials:

The selection of materials for the vehicle was made by the criteria of the extreme temperatures, metal strengths and weight. For the suspension a material was needed that was ductile but not too weak to sustain the torque applied by the wheel load. Aluminum, with its excellent strength to weight ratio, works well and is relatively light. It can withstand a torsional angle of 19.3 degrees which is acceptable for our purposes. Titanium, while having an even better strength to weight ratio, is too heavy and stiff for this application. Since aluminum has such excellent qualities it will also be used for the wheels, the hull and bin. The drive train poses a tough problem in terms of the weight constraint. Because of the wear involved and the dusty conditions much of the drive train will be nickel-plated aluminum and/or steel. This will increase the weight considerably but will protect against wear and any metal property changes due to heat accumulation. This will also reduce the size requirements of the total drive train. Lubrication appropriate for the conditions, such as Krytox 143A2 oil, will also be employed to prevent wear.

2.0 POWER SOURCE

2.1 Description:

The power source will be hydrogen-oxygen fuel cells. Fuel cells were chosen over batteries because they have higher energy density when power is needed over an extended period of time. Fuel cells can also be "refilled" be resupplying the fuel and oxidant sources.

The hydrogen-oxygen fuel cell is not the only fuel cell available. Chlorine-Magnesium, Chlorme-Zinc, Nickel-Cadmium and others were considered to be the most efficient and reliable at this stage of technology.

2.2 Design:

As shown in Figure 7, solid-state hydrogen-oxygen cell consists of a negative electrode (ne) and a positive electrode (pe) placed directly on each side of a solid electrolyte. The electrodes are 3-phase and are porous to gas. They contain both solid-state protonic conductors (SSPC) and catalyzing material. With an acid electrolytic (as in Fig. 8) the two electrode reactors may be expressed in a simplified way as:

ne: H_2 2 Had $2H^+ + 2e^-$ (1)

pe: $1/2\ O_2 + 2H^+ + 2e^-$ H_2O (2)

$H_2 + 1/2\ O_2$ H_2O (3)

At the (ne) or fuel electrode, one hydrogen molecule is absorbed and dissociated into electrons (e^-) that enter the electrode and hydrogen ions (H^+) that enter the electrolyte. At the (pe), which may be either a pure oxygen or an air electrode, oxygen molecules supplied at the electrode surface acquire electrons to form oxygen ions, which react with hydrogen ions to form the electrolyte. This completes the process of forming water, and the associated terminal cell voltage (Vc) as a function of the electric current (I) through the outer circuit.

This hydrogen-oxygen fuel is being used on the space shuttle and was used as a design guide because it is the most recent model which has been tested. The energy density of the fuel is 0.15 KW/Kg and the ARMS fuel cells will have a mass of 24 Kg. With a fuel efficiency of 1.2 KWh/Kg of fuel, 1,200 Kg of fuel are required for 12 days of operation.

3.0 PUMP

3.1 Description:

The shall be displaced from the bin onto the module by an "impeller pump". The pump shall work in a manner similar to a centrifugal fluid pump. The impeller will impart a maximum velocity of 5.015 m/s to the regolith which will then travel at an angle of 70 degrees through the directional chute of the casing (see Appendix 3).

The impeller and casing are designed to provide maximum efficiency on the basis of weight, reliability, and energy consumption.

The pump will operate intermittently due to the high flow rate of regolith produced by impeller speeds 4.635 m/s to 5.015 m/s and the low vehicle speed. The pump will be driven by a variable speed motor which will be attached to the impeller by the threaded socket. The socket will be 1/2 inch - 13 UNF.

3.2 Materials Selection:

The selection of materials for the pump is critical to the performance of the device, and, therefore, extra attention has been awarded to this process.

The pump casing or housing must be able to endure the continuous wear due to friction between the inside of the housing and the regolith. The inside of the housing shall be hard faced with a layer of tungsten carbide approximately 1.5 mm thick to provide maximum resistance of abrasion here. The inside of the casing will be hard faced to produce an extremely smooth and hard surface to resist wear due to the regolith. The casing itself and the impeller shall be titanium which is suitable for the harsh lunar environment. Titanium has excellent strength properties and can withstand the extremes of the lunar temperature range.

3.3 Lubrication and Bearings:

The regolith pump will utilize single row, deep groove antifriction bearings. The bearings will be lubricated by ion-plating a layer of lead film of approximately 0.2 micrometers on the bearings. The bearings can thus be protected from the cold welding effects and can warrant a lifetime of approximately 10^8 revolutions which results in about 86.9 days of service. Extra care should be taken to protect the lead from oxidation during on-ground testing and adjustments. Aside from this, the lead is an excellent source of bearing lubrication.

3.4 Weight, Mass, Inertia:

The total weight of the pump on earth is 30.75 Kg. Its moment of inertia about its axis is 0.1859 $Kg\text{-}m^2$.

3.5 Power Requirements:

To overcome the effects of inertia, approximately 14.28 N-m of torque will be needed to turn the impeller.

3.6 Failure:

The design of the pump lends itself to three failure modes. These include failure due to wear, overheating, and computer failure.

Extended use of the pump will in no doubt result in failure due to wear. After approximately 86.8 days of operation, bearing wear will be excessive and will most likely be the first incident of failure due to wear.

Overheating will be most prevalent when operating in warm to hot temperatures. Therefore, the vehicle will operate in moderately cold temperatures to suppress this mode of failure. The pump will also be painted with fuller white silicone paint to dissipate heat.

Failure due to computer is also possible due to the harsh environments of the lunar surface.

In any event, all parts of the design will be easily replaceable if failure occurs.

4.0 IMPELLER MOTOR

4.1 Description:

The motor to drive the impeller will be Sterracin Corporation model R9223-01 D.C. brushless torque motor (see casing Figure 9). The advantage of the brushless motor is the increased life expectancy by elimination of sliding brush contacts. Also, the brushless motor is capable of operating at higher speeds, and it has greater cooling efficiency since the wound member is mounted to a heat sink. Parameters given below:

Peak Torque:	20.4 N-m
Motor Constant:	2.42
Peak Power:	180 Watts
Temperature Rise:	2 C/Watts
No-Load Speed:	413 rad/s
Peak Current:	22.7 Amps
Voltage:	13 Volts

This motor was chosen because it is the most efficient motor to meet the torque and power requirements of the impeller. In addition, this motor has a frame framework already built in which makes it ideal for attachment to the impeller casing. Four beta-titanium rivets will be used to secure the motor to the casing (see Casing Calc. 1 for computation of force on rivets in Appendix 4).

4.2 Control:

The motor speed is variable and will be controlled by means of a variable resistor within the motor (see Figure 10). By changing the resistance, the motor speed will change accordingly.

4.3 Failure:

The only way the motor could fail is by requiring more torque than it is capable of producing. This should not happen since the maximum torque requirement necessary for the impeller is 17.43 N-m. The motor is capable of easily producing the maximum torque needed to drive the impeller.

5.0 BIN

5.1 Description:

The bin was designed to allow a funneling of the regolith towards the pump inlet in the rear. A lip in the front of the bin was added in an attempt to catch any thrown lunar soil which would otherwise be discarded under the vehicle. The height and width of the design were important so that regolith thrown from various distances would be collected. As such, the bin will be made to allow for regolith thrown as high as 55 degrees from 4 feet away or 71 degrees from a 2 foot distance. Ideally, the soil should be tossed at an angle greater than 54 degrees from close range to assure that it is properly obtained. Our optimum design has a height of 6.8 feet and a width of 6 feet (see Figure 11).

Bin designs were evaluated considering the following criteria:

1. Size: Must be large enough to catch regolith from a variety of angles and distances
2. Funneling: Must be capable of funneling regolith towards the pump opening.
3. Mounting: Must be able to mount to the existing frame design easily.
4. Weight: Having met the above criteria, the bin must be relatively light.

Thicknesses of the bin were selected on the basis of structural importance. The critical areas include the mounting plates on both sides of the bin, the pump mounting plate at the rear of the bin, and the bottom pan of the bin. Therefore, these areas will be made of 1/2 inch thick aluminum. All other non-critical areas of the bin shall be 1/16 inch thick aluminum.

5.2 Material:

The selection of material for the bin was important for strength and weight considerations. As such, titanium and aluminum were the two materials ideally suited for this design. Aluminum, with its high strength to weight ratio, was chosen on account of it being 1.7 times lighter than titanium. The weight of the bin will be equal to 293 pounds. Thus, while mot losing many of the benefits of titanium's strength, aluminum would save an average of $4.3 million at $22,000 per pound shipping weight. The bin's weight will also be advantageous in balancing the fuel cell weight in the rear of the ARMS vehicle.

6.0 SCHEME OF OPERATION

To adequately cover the module, a minimum of 30,000 ft^3 (see Appendix 6) of regolith is required. To allow for imperfect covering, a value of 40,000 ft^3 was chosen. The collection system is designed to dig 6 inches into the soil. This means that 80,000 ft^2 of the surface must be covered. The pattern chosen covers roughly 100,000 ft^2 to allow for obstacles and imperfect collection.

In order to simplify the controls and minimize the total distance traveled, a symmetrical path around the module was chosen. As shown in Figure 12, the path consists of straight lines connected by 45 degree turns. This pattern allows the device to operate in straight lines and make even turns.

The device will be controlled by an onboard computer with sensors that can determine the location of the module and obstacles in its path. The device will start out 260 feet away from and parallel to the module. It will then proceed to make 40 concentric laps around the module with each lap inside the previous one. This will bring the device to within 20 feet of the module and should cover the module. If it encounters obstacles, it will stop collection, go around the obstacle, and then resume the path and collection.

This pattern represents a simple solution for relatively ideal conditions. For an actual situation, a more complicated pattern would most likely be necessary to account for other modules and landscape features.

7.0 REGOLITH COLLECTION

In order not to damage either the module or the pump, the device must only throw particles of regolith no bigger than 1/2 inch in diameter (see Appendix 6 for calculations . Thus any collection system must both sort and collect regolith. This is accomplished by using a series of vertical disks fitted with fins (see Figure 13). When the disks are turned at appropriate angular velocities (200 - 600 rpm), the fins will throw the fine regolith while rocks will spin off the fins (see Appendix 6 for experimental results).

An arrangement of ten disks, with each disk 18 inches in diameter, was chosen to cover a width of 6 feet (see Figure 14).
The assembly consists of two shafts, each with five disks, angled at 30 degrees from the direction of the vehicle. These shafts will be connected to a center shaft by means of bevel gearing. A motor on the center shaft will power both shafts. Each disk will be fitted with four fins that are 1/2 inch in width (see Figure 13). The shafts will be supported by two bearings on each shaft.

To achieve an adequate volumetric flow rate, an angular velocity of 500 rpm was chosen. Assuming that the disks will dig in 6 inches, this gives a flow rate of 0.17 ft^3/sec which allows the vehicle to move at 0.06 ft/sec while collecting. The required torque on the shafts is 150 ft-lbs. To supply this a 1.5 hp (or 1.12 KW) dc motor was chosen. The motor is oversized to account for frictional losses.

The shafts and disks will be made of steel. Both the strength and the weight of the steel are necessary for the digging process. This assembly will be supported by an aluminum frame. The total weight of the assembly and frame will be 190 pounds.

The motor and gears will be covered by a sheet aluminum box. This box will help protect the mechanisms from dust and sunlight. The heat generated by the motor will be conducted away by the shafts and the disks.

Detailed calculations of volume, torque, and power as well as other design parameters are in the Appendix 6.

CONCLUSIONS AND RECOMMENDATIONS

This report proposes a device that will both gather the regolith and cover a module with it. The device described fulfills the objectives of the problem statement. We feel that this is a workable design that should be pursued further. A control system could be developed and design modifications could be made. The design makes use of existing technology and could be adapted for other purposes.

After the preliminary design, we have recognized different systems that merit consideration. These include: control systems that use a satellite data link; bury the module instead of covering it; use the device to perform other functions such as transportation, terrain exploration, and utility operations.

ACKNOWLEDGEMENTS

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Bob Porter, NASA Program Development (Huntsville, AL)

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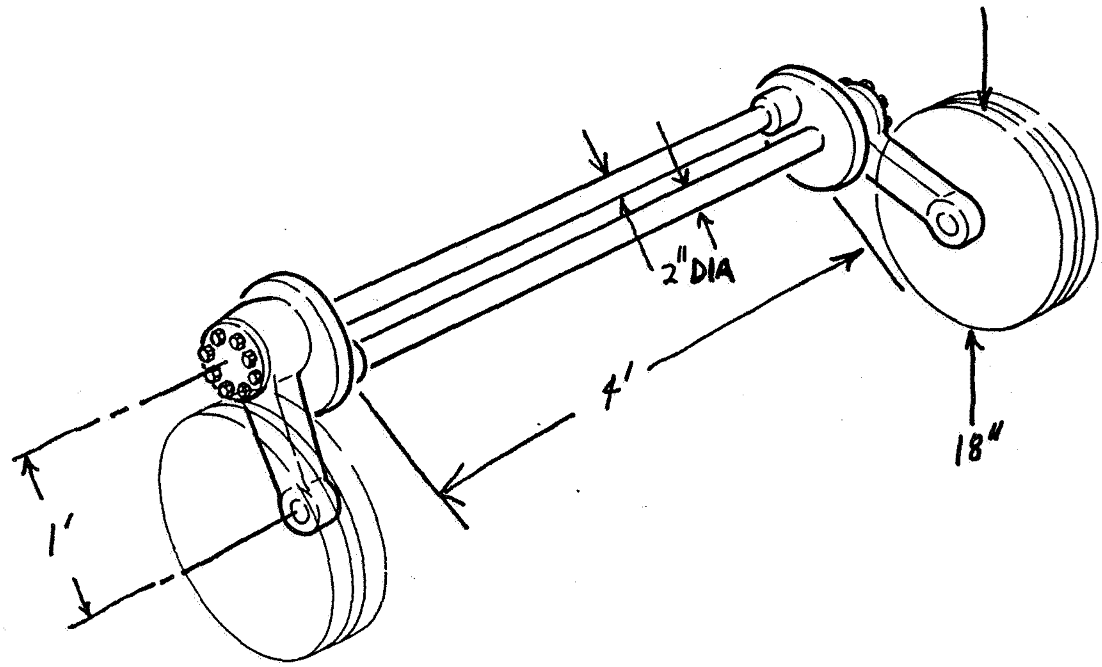


Fig. 2 Torsion bar layout

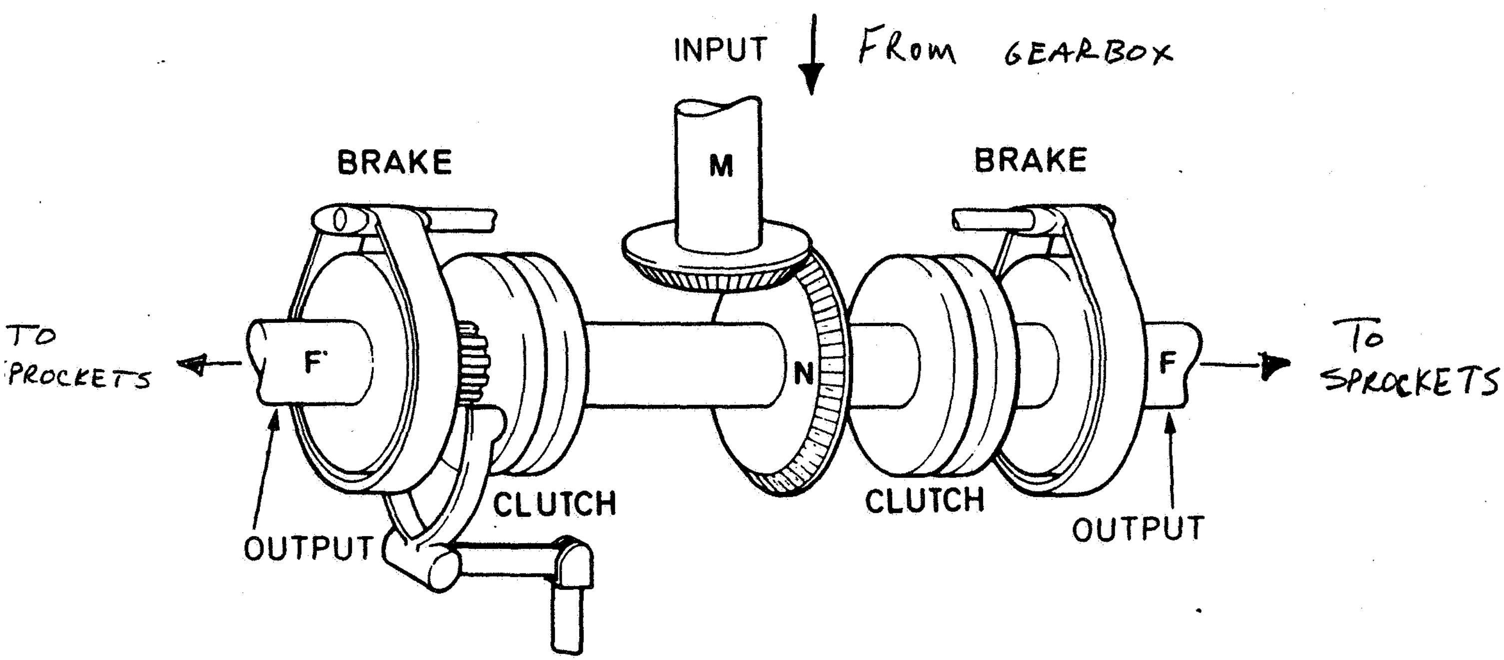


Fig. 3 Clutch-brake system

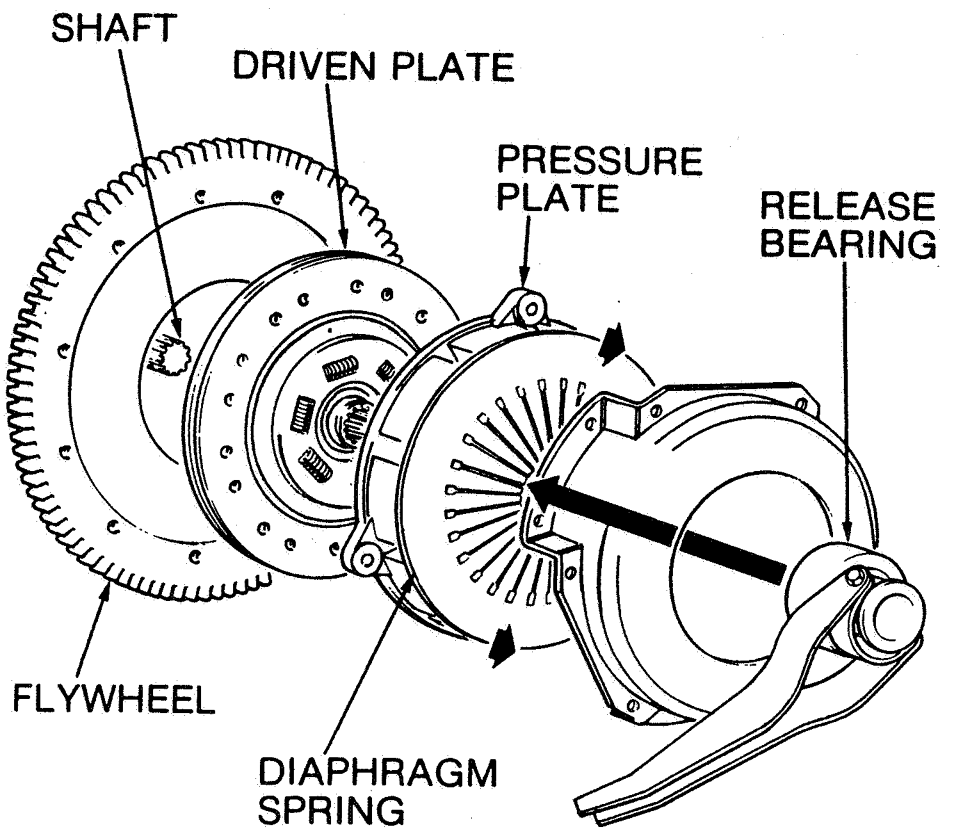


Fig. Friction clutch

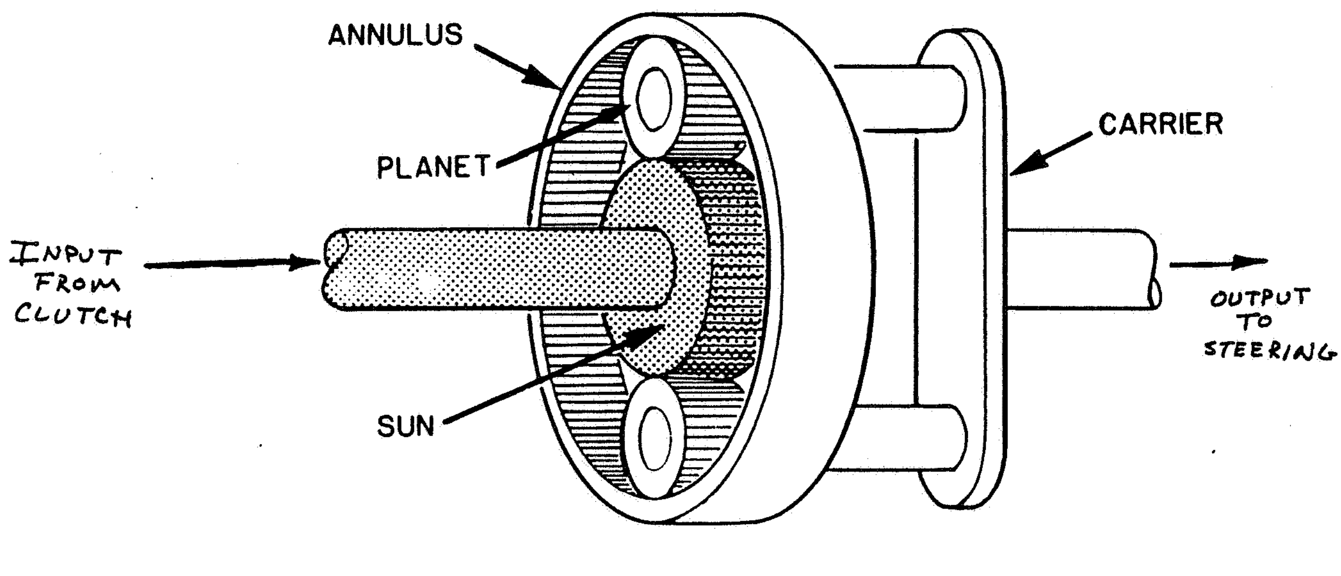


Fig. 4 Simple epicyclic

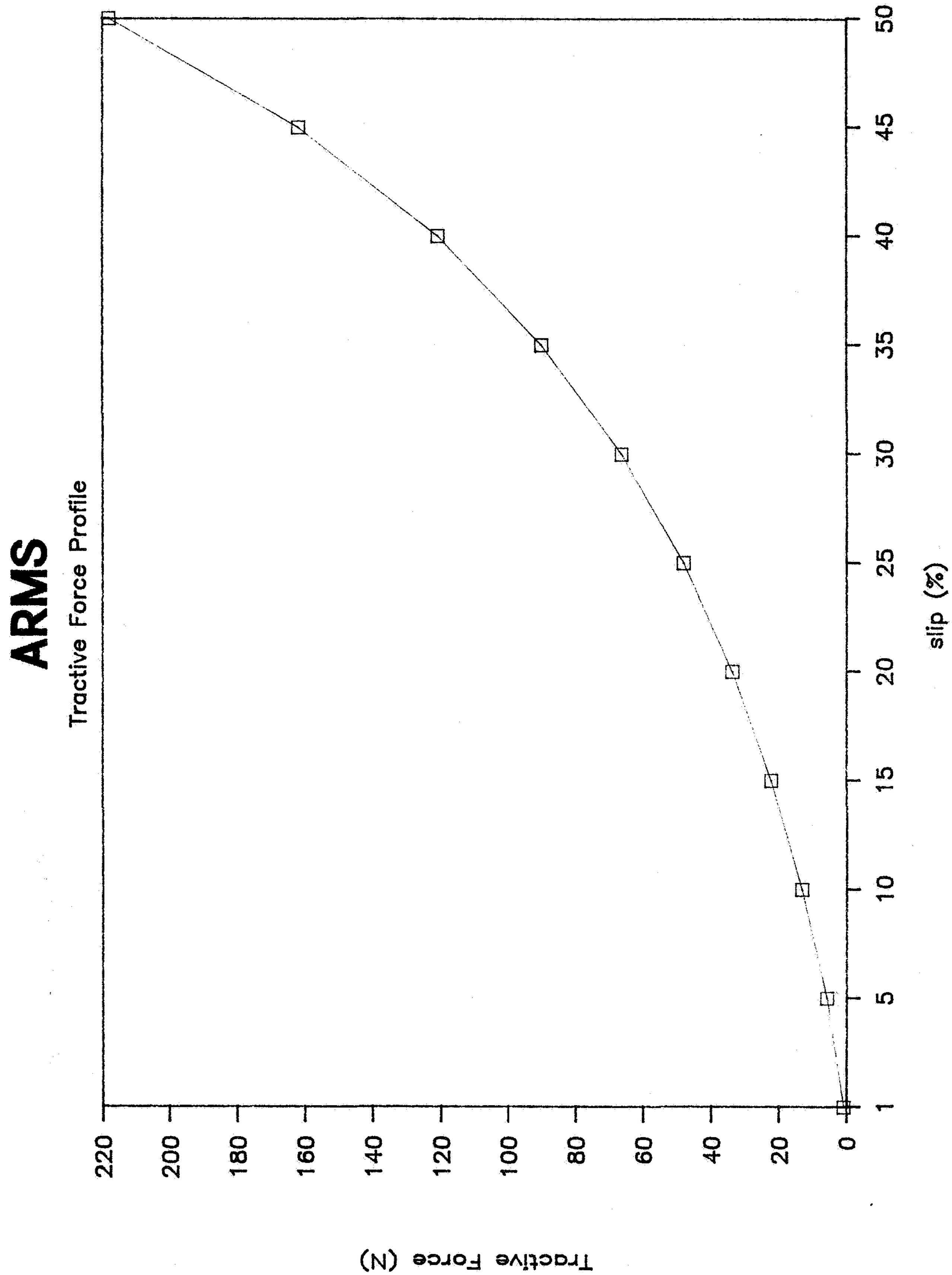

ARMS
Tractive Force Profile
Tractive Force (N)
220
200
180
160
140
120
100
80
60
40
20
0
1
5
10
15
20
25
30
35
40
45
50
slip (%)

Figure 5

26

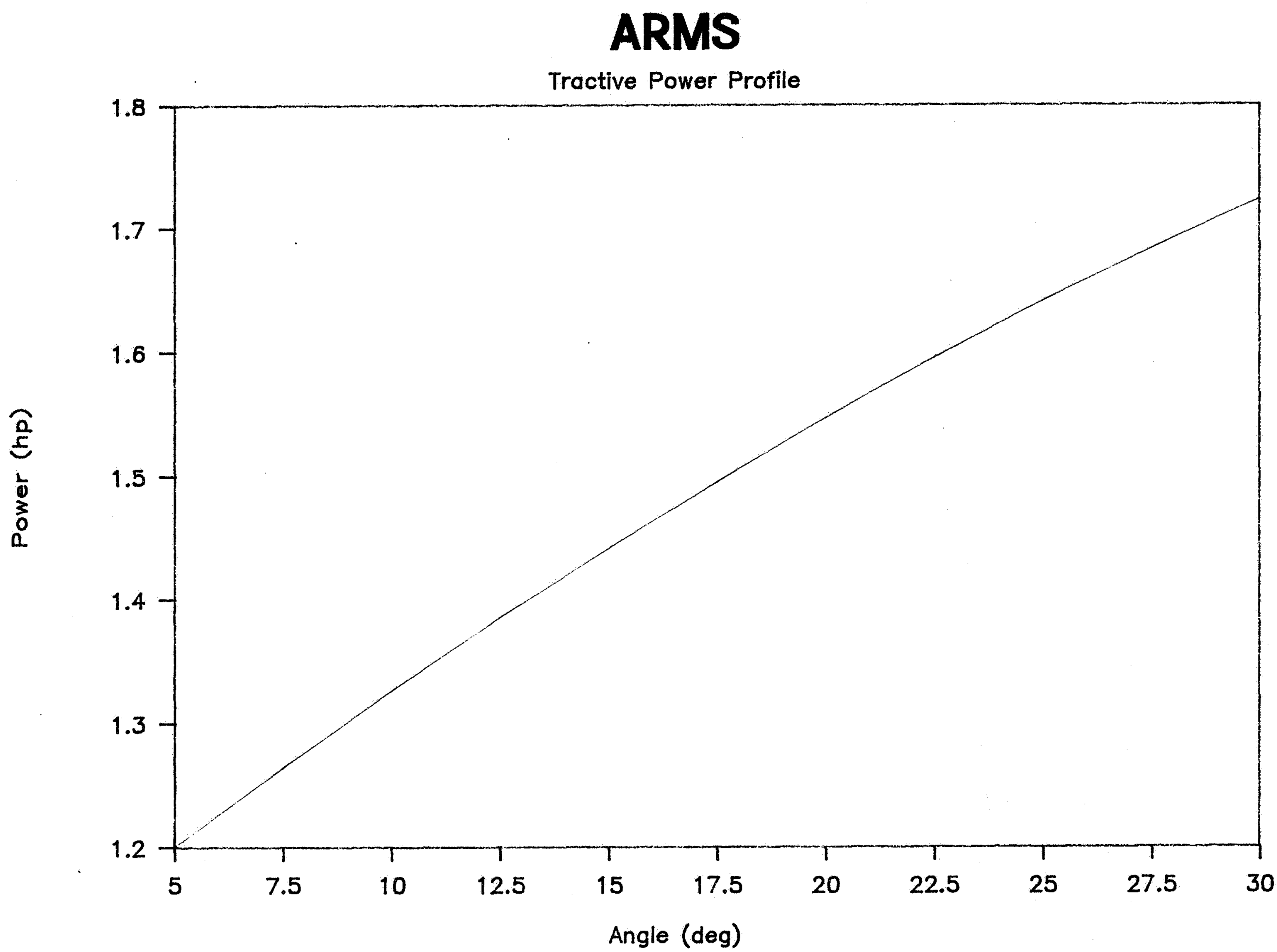
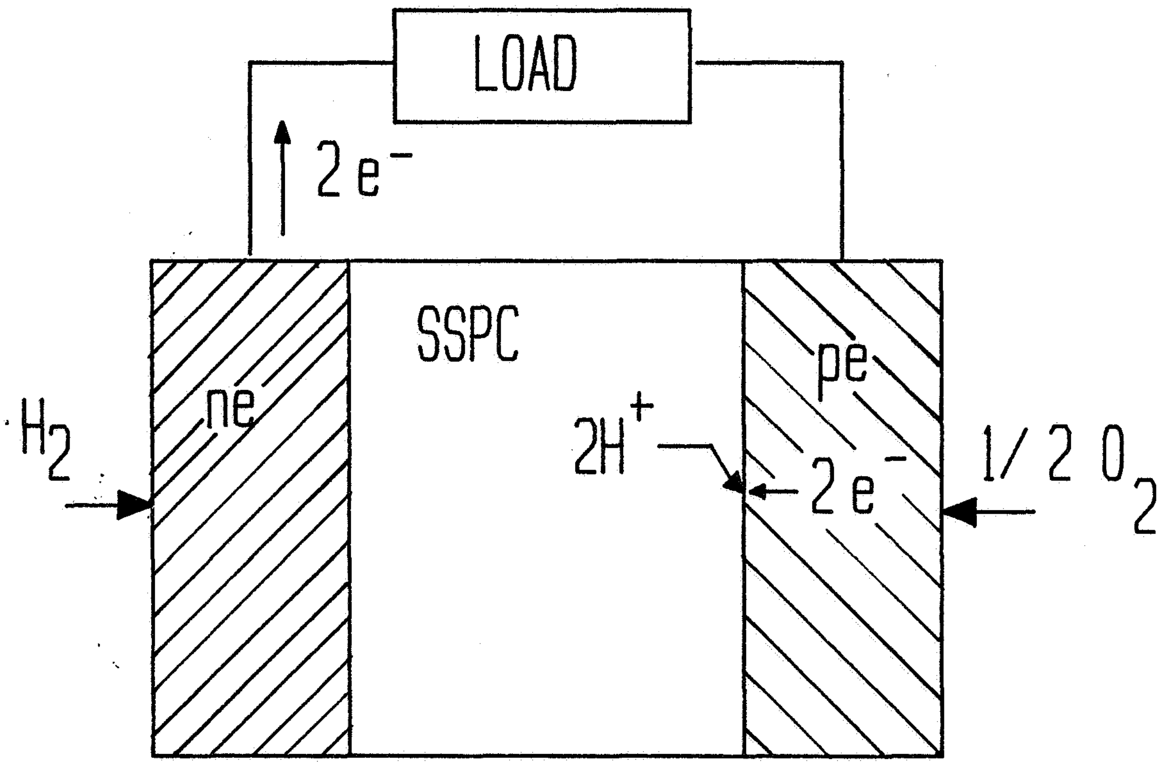

ARMS
Tractive Power Profile
Power (hp)
1.8
1.7
1.6
1.5
1.4
1.3
1.2
5
7.5
10
12.5
15
17.5
20
22.5
25
27.5
30
Angle (deg)

Figure 6



SOLID-STATE HYDROGEN-OXYGEN FUEL CELL
(SSPC - SOLID-STATE PROTONIC CONDUCTOR)

Figure 7

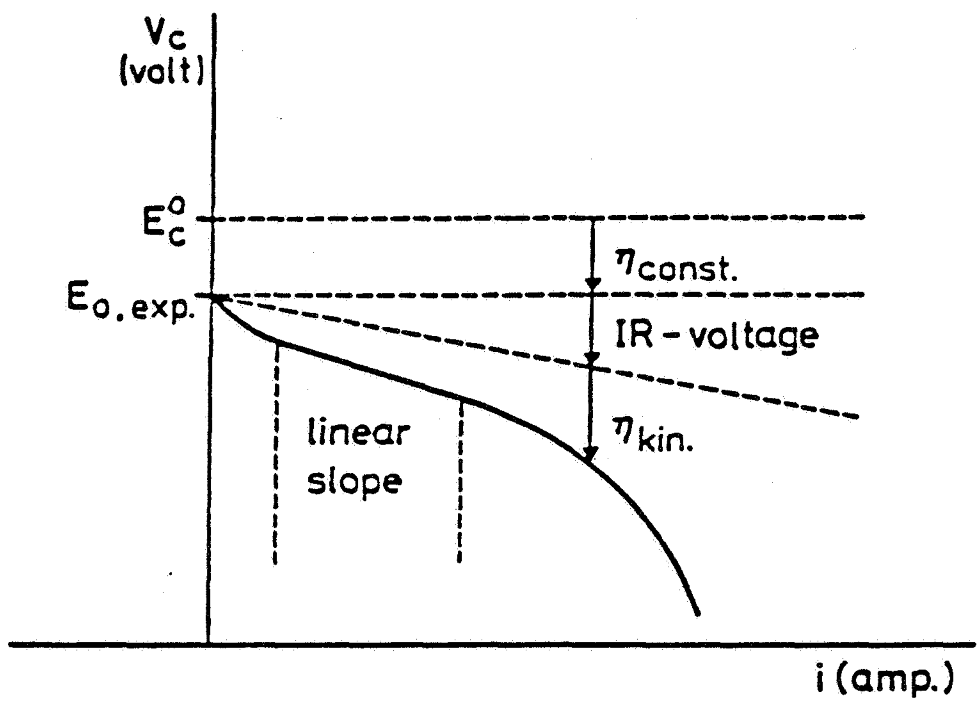
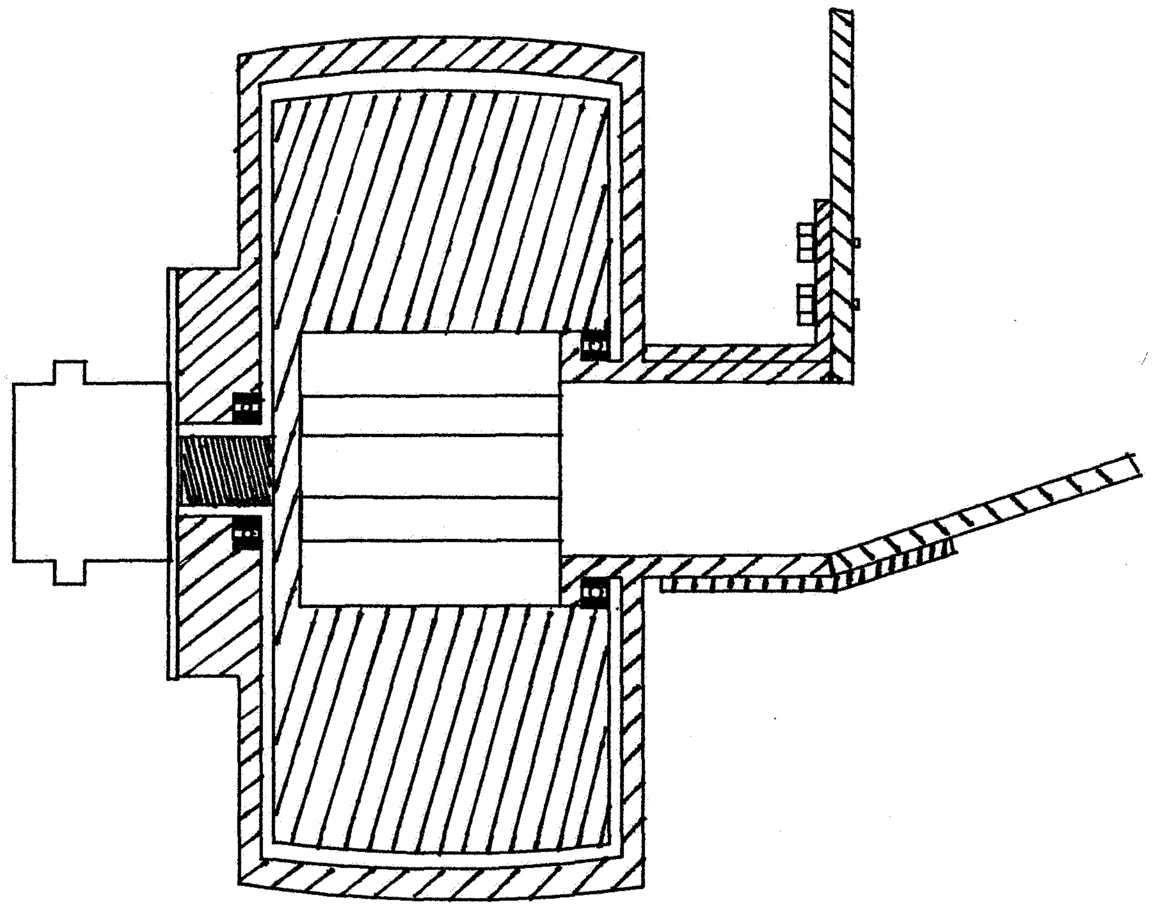


Figure 8. The cell voltage V_c of a hydrogen-oxygen acid-electrolyte cell as a function of the current i (from ref. 1, p. 162).



IMPELLER, CASING, AND MOTOR CROSS-VIEW

Figure 9

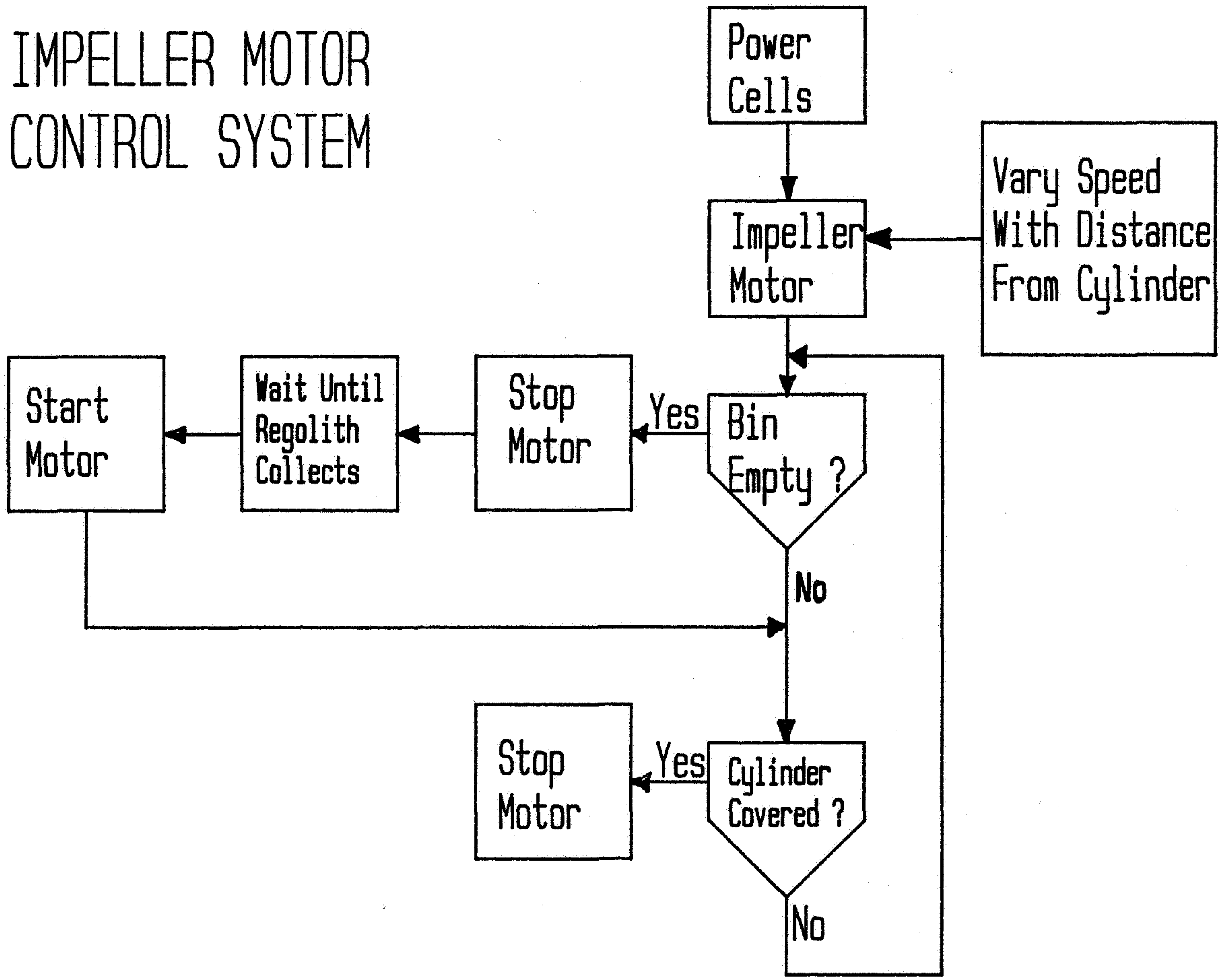

IMPELLER MOTOR
CONTROL SYSTEM
Power Cells
Impeller Motor
Vary Speed With Distance From Cylinder
Bin Empty ?
Yes
Stop Motor
Wait Until Regolith Collects
Start Motor
No
Cylinder Covered ?
Yes
Stop Motor
No

Figure 10

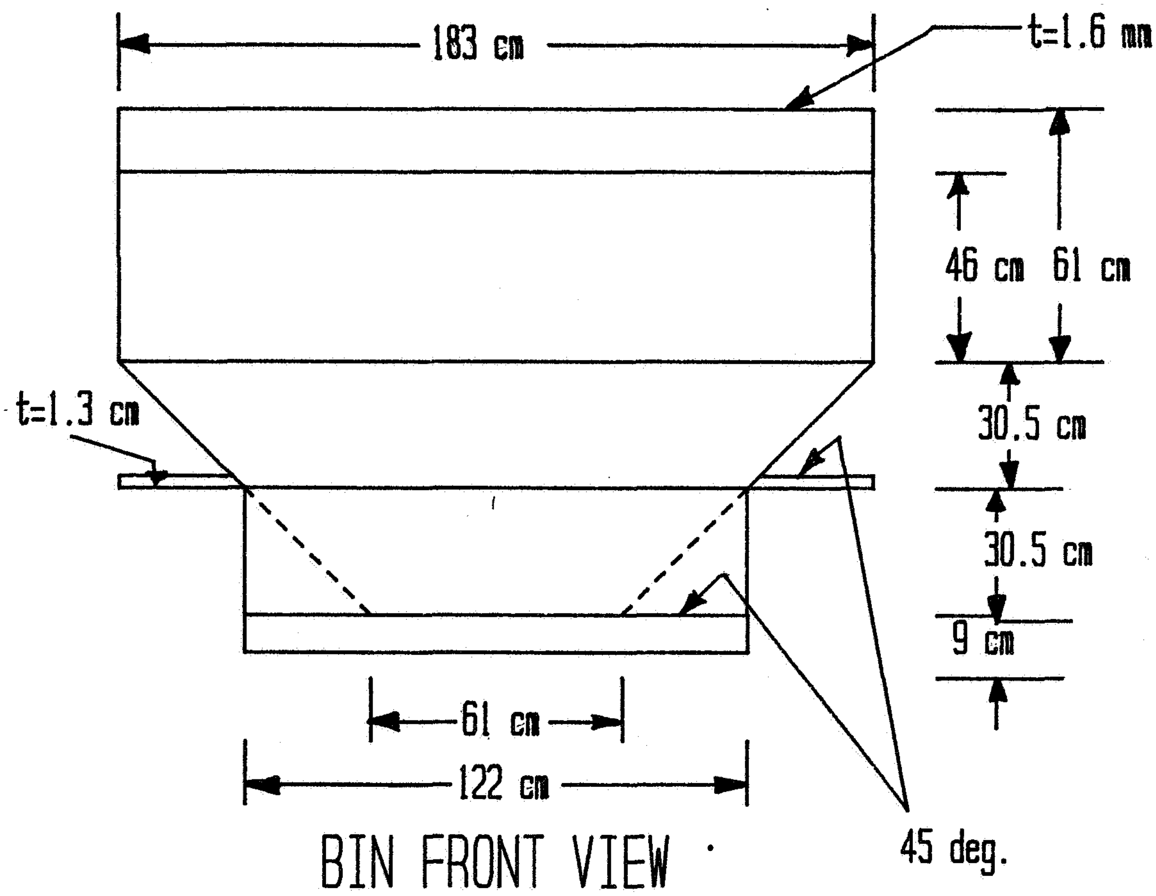

183 cm
t=1.6 mm
46 cm
61 cm
t=1.3 cm
30.5 cm
30.5 cm
9 cm
61 cm
122 cm
45 deg.
BIN FRONT VIEW

Figure 11

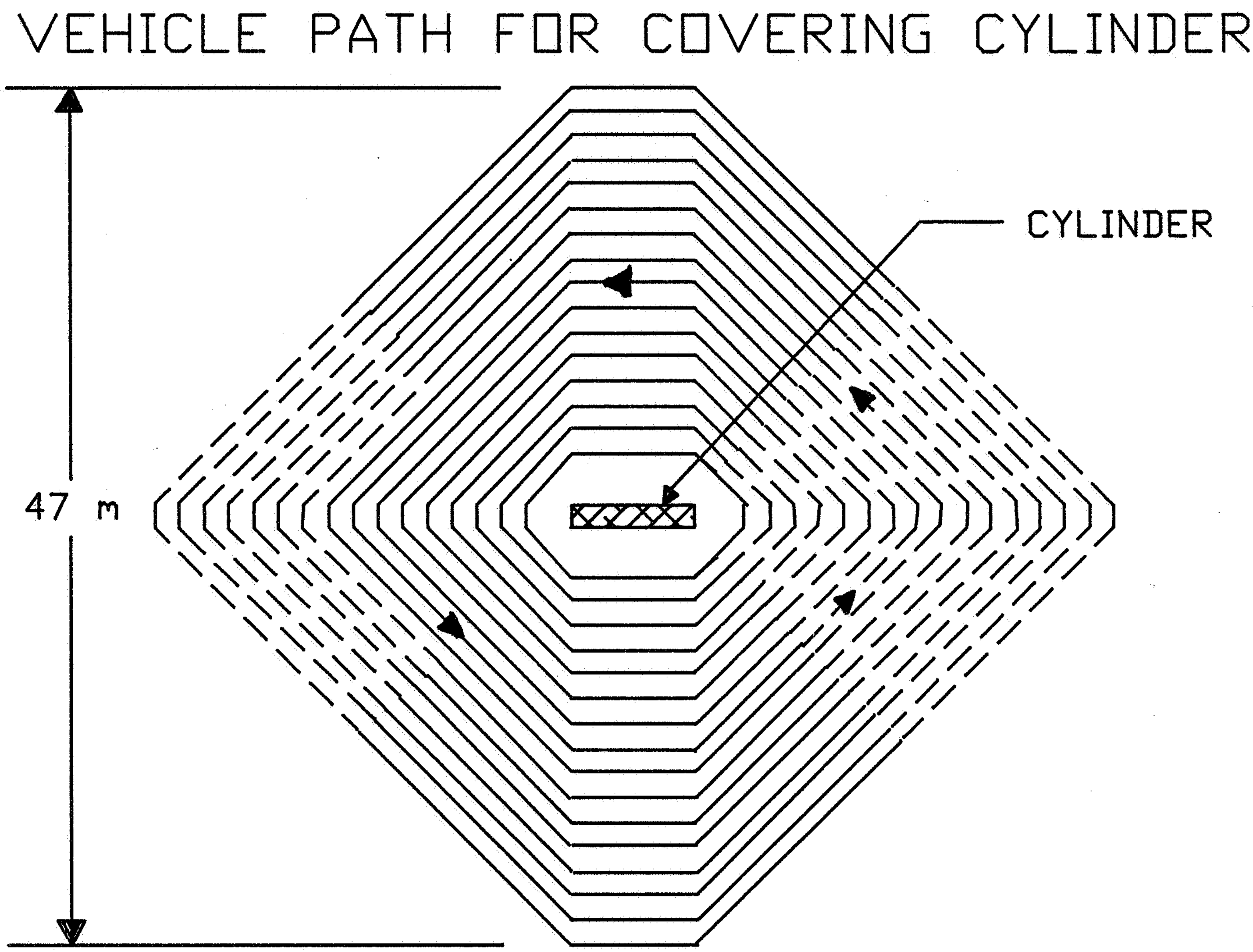


Figure 12

REGOLITH SORTING-THROWING DISKS

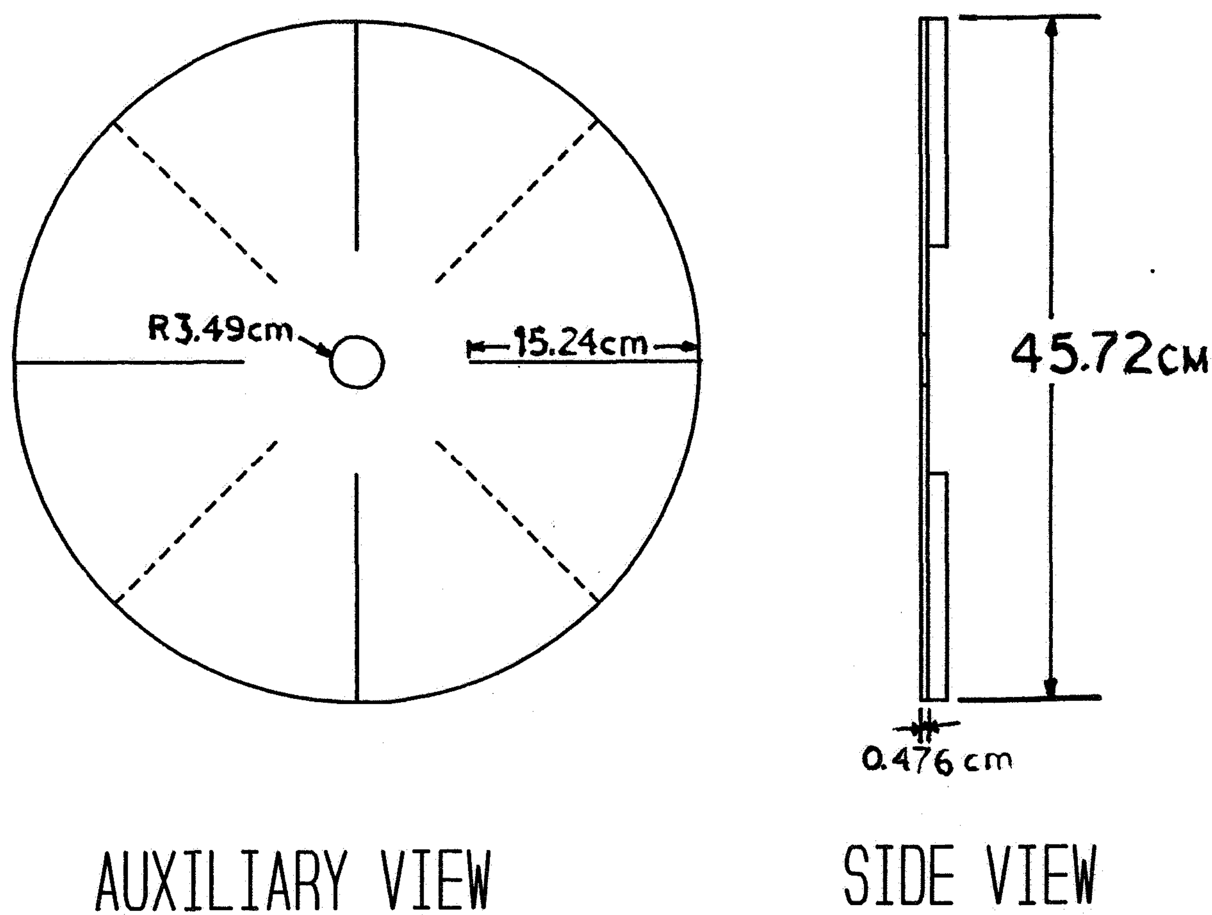
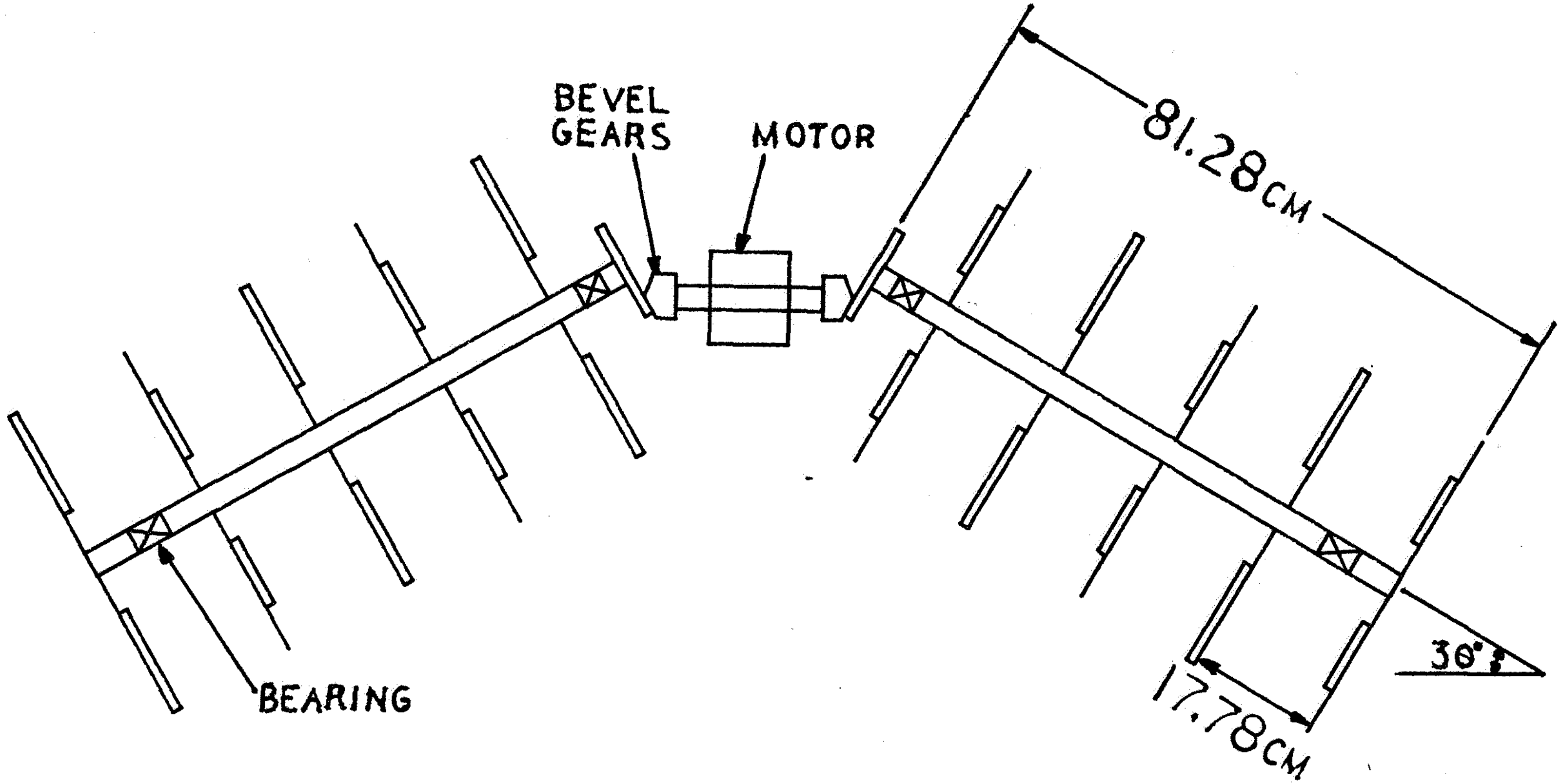


Figure 13



DISK ASSEMBLY TOP VIEW

Figure 14

Appendix 1

Vehicle

Book? → Reference?

1 ft = .3048 m

LUNAR TERRAIN - Possible BASE sites.

SAE
AG Engineers
Auburn, Lab

TERMS: See p. 127. Geology of the Moon

Areas favorable for Moon BASE SITE: MARE

Regolith: page 256

(1) Surveyor Program 1968b reported regolith depth of 4 Mare sites to be between 1 to 10 meters.
90% of particles (of this Regolith) were determined to be less than 1mm.
Assumption: Regolith layer thickens with time. ∴ Find oldest site.

(2) Three Orbiter Sites in these Regions: MARE TRANQUILLITATIS, OCEANUS PROCELLARUM and one site in SINUS MEDII found median Regolith thickness to be about 4.6 meters.
Northern SINUS MEDII and MARE IBRIUM recorded Regolith depths to be 7.5 meters.

(1) Shoemaker, Surveyor Program 1968b.
(2) Oberbeck and Quaide, 1968

Generally accepted Fact: MARIA formed shortly after Moon itself, ∴ oldest sites.

Apollo 12: landed MARE site, around 300 km south of Copernicus, in region in eastern part of Oceanus Procellarum.
relatively immature regolith, all crystalline. (Copy pix on Pp. 286, 87)

Apollo 11: landed in SW part of MARE TRANQUILLITATIS. To NORTH and South are rocky plains, Strewn-fields of Coarse blocks. LOTS of Craters. OLDER Regolith

Apollo 15: MARE MATERIAL site of Palus Putredinis, along southeastern rim of Imbrium basin. (Copy P.c. on P. 297)

Regolith: (copy Picture, p. 332) ← Apollo 12

Lunar Surface summary:

Vehicle will need to traverse small continuous boulders and craters (craters ½m — 5m wide, Boulders up to 3m).
majority of terrain features small 1-1½m craters and small ¼-½ m boulders. optimum design → treads.

[18 in tread, ~~1~~ ft clearance, 4 ft wide inside to inside)

LUNAR BASE SITE: MARE TRANQUILLITATIS (Close to eastern highlands)

- Very old regolith, meaning deeper. Regolith thickens with time.
- 90% of particles (of regolith) were determined to be less than 1 mm thickness.
- Median Regolith thickness was 4.6 meters (or 15.1 ft) (see picture, Apollo 11 landing.)

- Craters of sizes ranging from centimeters to yards will be encountered. Rocks of similar sizes also.

[Taken from book Geology of the Moon, references being Shoemaker (Surveyor ~~1968~~b) and Oberbeck and Quaide (1968).]

Initial Concept:

1. Wheels - traction problems around obstacles. Not stable platform and everything
2. ☒ tracks - Good traction, could go over anything
3. Walker - not stable enough, auger/thresher would not be close to ground much.
4. Rope and Pulley - restricts motion and manuverability. Not as versatile designing for future also.

suspensions:

(X) depending on design overall (X)

1. Bogey Suspension: simple system, spring operated. Both sides independent of each other. good for slow speeds. Because it's external to hull it is easy to replace and there are no shafts connecting to bogeys on other side of vehicle. (see page 118, Fig 6.15 and page 119, Fig 6.16). —TOO HEAVY—

2. Torsion bar: simple, light, relatively inexpensive. wheel load is made to apply a torque to a circular section steel bar whose twist in response provides the required compliance. considerable problems in repair. raised hull height required too. (pp. 121-122) ~~impractical due to overall design~~

3. Hydrogas: operate at low pressures of approx. 7 bar static but are bulky, vulnerable and unsuited for fighting vehicles. operating at 100 bar static, 500 bar static at bumps. Sealing of piston is problem. (Hydraulics on moon? too hard or easier design?) This system could be usefull in raising ground clearence (or lowering) when nessesary. Size considerations on the hull size make it unpractible as well as difficult maintenance. Requires too much maintenance.

4. Coil spring: externally mounted and bulky. Because the stress that can be employed in this system will be reduced by 25% of that for the torsion bar. This makes it considerably heavier for a given resilience. No longer used on tracked vehicles. Not needed as much on the Moon. Smaller springs

(3) Hydrokinetic Coupling: engine drives a vanned wheel, the pump, which in turn gives whirling motion to oil. This oil drives the vanned turbine wheel which drives the Gearbox. Torque taken from the oil by the output is equal and opposite to that provided by the input. there is some slip so output will not equal input. smooth drive. good efficiency.

(4) Hydrokinetic Converter/coupling: (also called Torque Converter). a stator is added to both pump and turbine to greatly improve output torque. Provides both an automatic clutch at starting from rest and a degree of "lowering" of the gear in use, automatically w/o changing gear! This means it can act as Perhaps 1½ extra gears. If cooled adequately it can allow for continuous low speed movement. Usually combined with an automatic-change epicyclic gearbox. Greatest value in low gears.

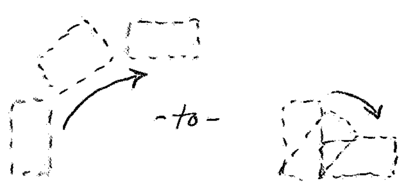
DIFFERENTIALS

(1) Bevel: most common, drives the output shafts at equal speeds, or unequal speeds depending on the situation. relatively simple and compact.

(X) (2) Differential Lock: Locks built into a differential, automatic control possible, enables vehicle to maintain traction on at least one side at all times, Crucial for automated use, especially total automation like on moon.

STEERING:

(X) (1) Clutch-BRAKE steering: Skid steering is most common, disengaging one side's clutch, turning the vehicle. sharpness of the turn depends on the length to width (L/c) ratio. (See dimensions) Variable types of turns can be achieved. Partial braking creates excess heat which could be regenerated to heat vehicle systems during cold lunar periods.



(2) Twin epicyclic: turns vehicle by same concept as the Bevel differential, larger than clutch-Brake.

(3) Merritt Double Differential: changes added to epicyclic above, even more steering finesse at higher speeds. high Speed not needed.

(4) Hydrostatic Steer: Another variant, for more finesse steering at higher speeds. this too, is too big.

wheels → $\frac{1}{3}$ the contact area. ∴ 3 times the NGP.

CRITICAL DIMENSIONS:

nominal ground pressure $NGP = \dfrac{\text{Vehicle wt.}}{\text{area of tracks in contact with ground}}$

Length to width ratio: $\boxed{L/C \simeq 1.5 \text{ to } 1.8}$ PATH: (inside) 4-[illegible] ft.

Hull height: depends on engine and suspensions utilized.
Hull length: L/C criterion.
Hull width: engine, drive train, thresher/auger device.

TREAD DESIGN: ~1/3 the ground pressure of wheeled Vehicles.

- for easier turning a low L/C ratio is needed.

① Light-weight links, rubber or comparable material, with metal rods for links (Rubber Bushed).

② Light wire mesh, reinforced with Chevrons attached for traction. Similar to Lunar rover tires.

③ Continuous Rubber track - Rubber won't hold up. (?)

- avoid losing tracks when turning by placing metal "horns" on track (inside) to run through guide or road wheels.
- sharp aggressive edge, or grouser, is needed for traction ⇨ chevrons.

Obstacle avoiding methods:

① stop-motion sensor: when the vehicle hits something or gets stuck, it will automatically go in reverse to escape, turn and go on.

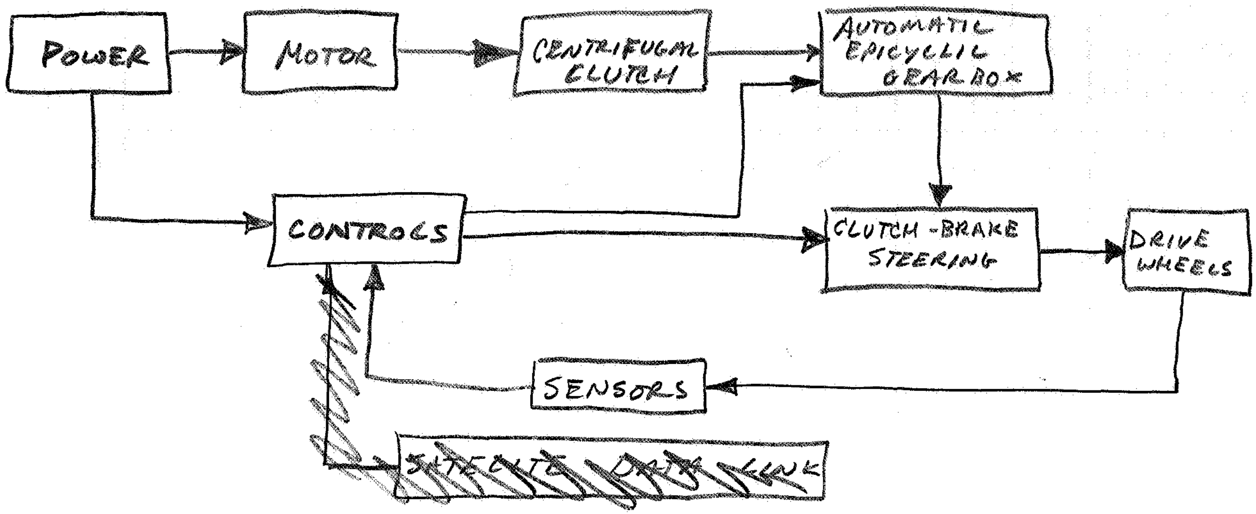
② some kind of laser range finder or infrared sensor to "see" large obstacles to avoid them. (TOO DUSTY to use this)

③ Cow-catcher shaped wheel/thresher assembly. with easy depression/spring sensors for sensing Craters or rocks (both)

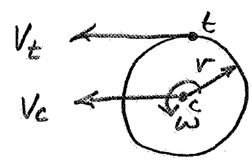
④ SATELITE DATA LINK TO Vehicle - Satelite sees terrain (very closely) and automatically guides Vehicle.

- fuel cells
- plug into power drone.

Vehicle DRIVE-TRAIN Flowchart:



— CLUTCHES AND GEAR CHANGER ARE HERMETICALLY SEALED FOR CONTROLED ENVIRONMENT OPERATION.



r = sprocket radius = 6" = .1524 m

$V_c = r\omega$

$T = I\alpha$ $\quad I = \frac{mR^2}{2} = \frac{(400\,kg)(.1524\,m)^2}{2}$

$I = 4.645\ kg\cdot m^2$

$g = \frac{1}{6}(9.81\ m/sec^2) = 1.635\ m/sec^2$

SPEED : $\frac{5\ miles}{Hour} \times \frac{1\ Hour}{3600\ sec} \times \frac{5280\ ft}{1\ mi} \times \frac{1\ m}{3.2808\ ft} = 2.24\ m/sec$

V_c (mi/hr)	m/sec	V_t (m/sec)	ω (rad/sec)	α (rad/sec²)	Torque (N-m)	rpm	Power (hp)
7	3.14	6.28	20.6	10	46.45	197	1.284
6	2.68	5.36	17.6	8.8	40.88	168	.9642
5	2.24	4.45	14.6	7.3	33.91	139	.6636
4	1.80	3.60	11.8	5.9	27.41	113	.4334
3	1.34	2.68	8.8	4.4	20.44	84	.2416
2	.884	1.77	5.8	2.9	13.47	55	.1054
1	0.46	0.91	3.0	1.5	6.97	29	.028

Power = $T\omega$

V_t → track speed.
V_c → Vehicle speed.

$$V_t = V_c + (\omega_c \times r_{tc})$$
$$= 3.14\ m/sec\ (\hat{i}) + (20.6\ \hat{k}\ rad/sec \times -.1524\,m)$$
$$= 6.28\ m/sec \quad (\text{or } 2V_c)$$

Angular momentum = $L = I\omega$
Net torque needed to move vehicle,

$$T = I\alpha = \left(\frac{mr^2}{2}\right)(\alpha) = (4.645\ kg\cdot m^2)(10\ \tfrac{rad}{sec^2})$$
$$= 46.45\ N\text{-}m$$

$$a = a_c + \alpha \times r_{ct} - \omega^2 r_{ct} \qquad a_c = r\alpha$$

$$a = r\alpha + \alpha \times r_{ct} - \omega^2 r_{ct}$$
(4 mi/hr) $a = (.1524\,m)(5.9\ rad/sec^2) + (5.9\ rad/sec^2\ \hat{k} \times -.1524\,m\,\hat{j}) - (11.8\ \tfrac{rad}{sec})^2(-.1524\,m\,\hat{j})$

$$= 23.1\ m/sec^2$$

CLUTCH-BRAKE : output torque $6.97 < T < 46.45$ (N-m)
wheel speed $29 < n < 197$ (rpm)
horsepower $.028 <$ Power < 1.284 (hp)

pressure angle $\phi = 20°$

$\tan\gamma = \frac{N_P}{N_G}$ $\quad N_p = d_p P \quad N_G = d_G P$ $\quad N_p$ = teeth on Pinion, N_G = teeth on Gear

$\boxed{r_1\omega_1 = r_2\omega_2}$ $\qquad T = r_{av}\omega_t$

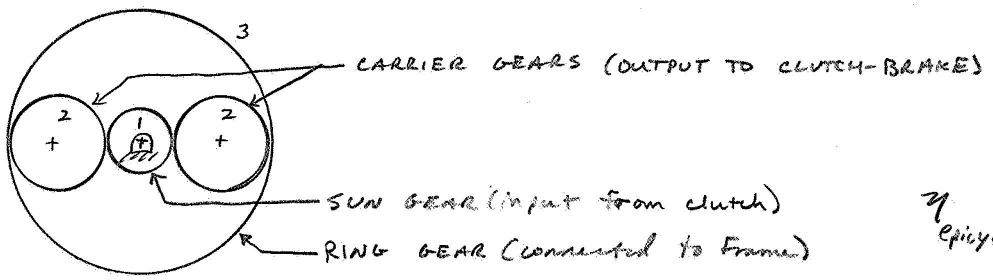
We want input ω_1 to be same as our output ω_2 (100% power transmission) (Next).

for 100% transmission $v_1 = v_2$. Expect an efficiency of only 90% though. ∴ $n_{2_{max}} = 197$ rpm , $n_{1max} = 197/.9 = 219$ rpm and

max. Torque of $T_{max} = 51.61$ N-m

Epicyclic gears:

Teeth ratios : $\frac{G1}{G2} = \frac{20}{30} = \frac{2}{3}$



$\eta_{epicyclic} = .70$

Torque ratio

$$\frac{T_{out}}{T_{in}} = \eta \cdot \frac{n_{in}}{n_{out}} \qquad n_{out} = 219 \text{ rpm} \quad T_{out} = 51.61 \text{ N.m}$$

Working backwards for the input speed gives:

$$n_{in} T_{in} = \frac{(51.61 \text{ N.m})(219 \text{ rpm})}{\eta = .70} = 16146.56 \text{ rpm.N-m}$$

Knowing that $n_2 = n_{out} = 219$ rpm

$$\frac{n_1}{n_2} = \frac{3}{2} \qquad n_1 = 328 \text{ rpm}$$

And $T_{in} = 16146.56 \text{ rpm.N.m} / 328 \text{ rpm} = 49.15$ N.m

CENTRIFUGAL CLUTCH - Assume 90% efficient
∴ input torque to clutch = 54.6 N.m

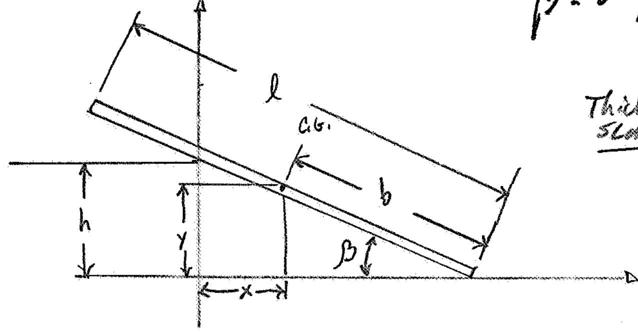
∴ Motor operates at $n = 365$ rpm and $T = 54.6$ N.m
$\omega = 38.22$ rad/sec

∴ Power $= T\omega = 2087.34$ N.m/sec $= \underline{2.8 \text{ hp}}$

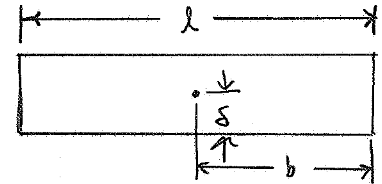
Negotiating Vertical steps

β = angle between Vehicle and ground.

Thin slab.



Thick SLAB



To negotiate the step, the C.G. x value must go negative. If $x > 0$ for all β the rod will not make it.

C.G. equation path: ① $x = (h/\tan\beta) - b\cos\beta$

② $y = b\sin\beta$

controlling parameters : h & b

For a thick slab of width 2δ, the equations of the C.G. path is:

③ $x = (h/\tan\beta) - b\cos\beta + \delta\sin\beta$

④ $y = b\sin\beta + \delta\cos\beta$

$[\sin^{-1} h/l \leq \beta \leq \pi/2]$

-From this the maximum step that a track layer can negotiate is determined using Parameters b and δ.

From equation ③, limiting case $x = 0$ you get :

⑤ $(h/\sin\beta) + \delta\tan\beta = b$

and then

⑥ $(h/b) = \sin\beta - ([\delta/b][1/\cos\beta] - \cos\beta)$

⑦ $h_{max} = b\sin\beta - (\delta/\cos\beta - b\cos\beta)$

to Find β, differentiate eq ⑥ you get :

⑧ $b/\delta = \tan\beta(2 + \tan^2\beta)$

- find the β for eq. ⑧ from Vehicle Geometry (β_{min}?)
- Substitute into eq. ⑦ to get h_{max}, largest step.

~~→ make one on energraphics (after making a table)~~

~~$b/\delta = 3.0$ or: $\delta = 1.5$ ft, $b = 4.5$ ft, $l = 9.0$ ft~~

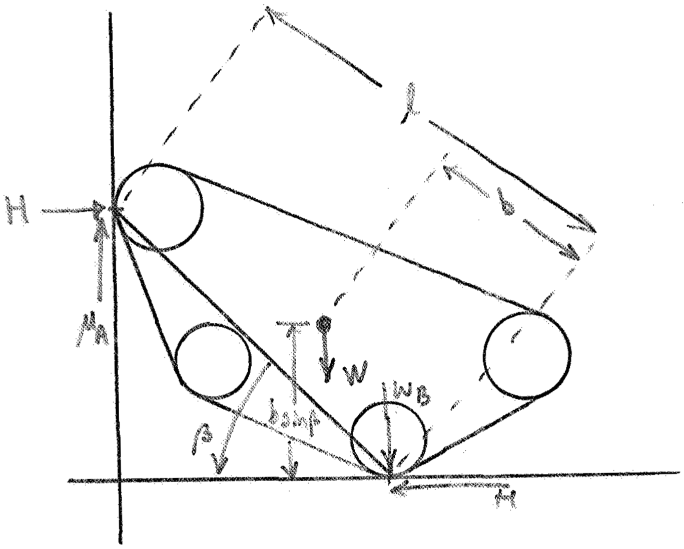
~~$h_{max} = 4.24$ ft (5 ft. high Vehicle, 9 ft long)~~

β	b/δ	δ	b	l	h_{max}	c
5						
10						
15						
20						
25						
30	1347	3.34	4.5	9.0	2.29	6/5

~~[set up chart of design values]~~

~~Use it to find ultimate width, c and overall dimensions]~~

Equilibrium of Forces



eq. of Moments $(H = \mu_B W_B)$

(1) $W(l-b)\cos\beta + H l \sin\beta - W_B l \cos\beta = 0$

Eq. of Vertical forces

(2) $W - (\mu_A H + W_B) = 0$

(3) $W = \mu_A \mu_B W_B + W_B = (1+\mu_A\mu_B) W_B$

$$\mu_A = \frac{1}{\mu_B}\left[\left(W/W_B\right) - 1\right]$$

using eq. (1) you get $W/W_B = \dfrac{l}{(l-b)(1-\mu_B \tan\beta)}$

and (4) $\mu_A = 1/\mu_B \left[\left(l/(l-b)\right)(1-\mu_B \tan\beta)\right] - 1$

$l = 2b$ then $\mu_A = \dfrac{1}{\mu_B} - 2\tan\beta$

or

$\mu_B = (\mu_A + 2\tan\beta)^{-1}$

$.577 < \mu_A < 1.19$ (regolith $\mu_B = .649$)

$\tan\beta = \frac{1}{2}\left[\frac{1}{\mu_B} - \mu_A\right]$ $\beta_{min} = 19.31^\circ$

The required μ_A for the vehicle to overcome the obstacle is

$\mu_A = \mu_B \, b/l$ our vehicle $b = \frac{1}{2} l$

$\mu_A = (.649)(1/2)$

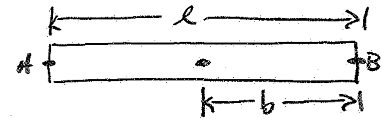
$\mu_A = .3245$

The vehicle will not be able to climb the obstacle if the obstacle μ is:

$$\boxed{\mu \leq \mu_A}$$

- otherwise the vehicle will simply slip against it.

SLIP RATIOS AT CONTACT POINTS.



Assume Velocity at pt. B = V_B
Finding V_A:

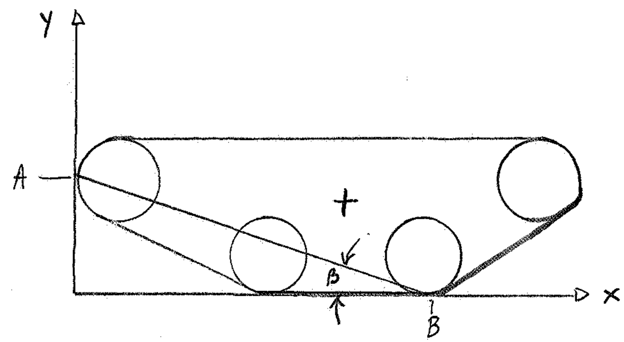
β = for this case is only angle shown. Not relative to ground.

I = instantaneous center of motion
$\omega = (V_B / l \sin\beta)$

$$① \; V_A = (V_B / l \sin\beta)\, l \cos\beta = V_B \cot\beta$$

since $V_A = V_B \cot\beta$, $V_A \to \infty$ would be needed with $\beta = 0$, However β is never 0 with vehicles.

V_t = track Velocity.



β_{min} for a tracked vehicle.

for $\beta < 45°$ $V_A \gg V_B$ for $\beta > 45°$ the situation is reversed.

two different slip values are required:

$$② \; i_B = (V_T - V_B)/V_T = (1 - V_B/V_T) \quad ③ \; V_B = V_T(1 - i_B)$$

$$④ \; i_A = (V_T - V_A)/V_T = (V_T - V_B \cot\beta)/V_T = 1 - (V_B \cot\beta / V_t)$$

$$(1 - i_B) = (1 - i_A)\tan\beta$$
$$(1 - i_B)/(1 - i_A) = \tan\beta$$

$$⑤ \; i_B = 1 - (1 - i_A)\tan\beta$$
$$⑥ \; i_A = 1 - (1 - i_B)\cot\beta$$

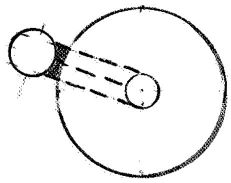
for $\beta = 45°$ $\left.\begin{array}{l} i_B = i_A \\ i_A = i_B \end{array}\right\}$ Slip is SAME.

- since in most cases the vertical wall or obstacle is firmer than the soil covering the approach it is likely that i_B will be greater than i_A. When β is more than 45°, i_B defined by the kinematics of the problem is less than i_A. One should then expect difficulty when the height of the obstacle requires a β of 45 degrees or greater.

$\pm 270°F$ →
$\pm 130°C$

Torsion bars

ROAD WHEEL = 18"
MOMENT ARM = 12"



$T = Fd$ F = Force pushing wheel up.
d = moment arm.

Torsion bar equations:

$$\theta = \frac{Tl}{GJ}$$

T = Torque, G = Modulus of elasticity
l = length, J = polar moment of inertia $\left(J = \frac{\pi d^4}{32}\right)$

yield strength in shear : $S_{sy} = .5\,S_y$ (conservative)
or
$S_{sy} = .577\,S_y$

Materials:

Aluminum $\rho = 2.70\ g/cm^3$ (1/3 of steel)

— Poor wear resistance (soft metal)

These two have similar wt/strength ratios at extreme temperature

① 3003-H18 $S_y = 27{,}000$ psi (Architectural uses)
(Aluminum-manganese alloy)
— Not aged, wrought alloy. Cold worked.

② ② 2024-T4 $S_y = 47{,}000$ psi %E = 20 (too Ductile)
4032-T6 $S_y = 46{,}000$ psi %E = 9 (too low solvus Temp)
① 7075-T6 $S_y = 73{,}000$ psi %E = 11 (tiertiary alloy)
(Al-Mg-Zn)

Titanium $\rho = 4.505\ g/cm^3$

Beta Titanium alloys (13% V, 11% Cr, 3% Al) (Vanadium)
$S_y = 176000$ psi %E = 5

Compare yield strength-to-wt ratio of Beta-Ti to 7075-T6 Al:

$$Ti = \frac{176000\ psi}{4.505\ g/cm^3} = \frac{176000\ psi}{.163\ lb/in^3} = 10.8 \times 10^5\ in$$

$$Al = \frac{73000\ psi}{2.7\ g/cm^3} = \frac{73000}{.096\ lb/in^3} = 7.6 \times 10^5\ in$$

* Titanium Torsion bars.

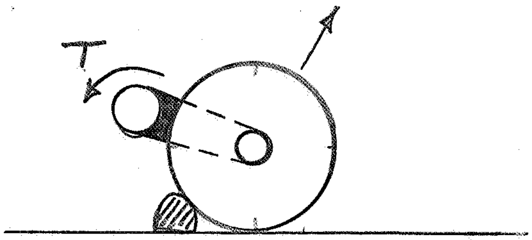
Beta Titanium : $\rho = .163 \ \frac{lb}{in^3}$ $S_y = 176{,}000$ psi

Torsional yield strength = $.5(S_y) = .5(176 \text{ kpsi}) = 88$ kpsi

Torsion bar area : $2\left(\frac{\pi}{4}(2'')^2\right)$
$= 6.28 \ in^2$

$\sigma = \frac{F}{A}$

$\sigma = \frac{S_{sy}}{n}$



FACTOR of SAFETY let $n = 1.5$ (?)

$\sigma = \frac{88k \text{ psi}}{1.5}$

$\sigma_{MAX} = \underline{58.7 \text{ kpsi}}$

$\therefore \ F = (58.7 \text{kpsi})(2\pi \ in^2)$

$F_{max} = 368.8$ kips.

$\begin{bmatrix} G = 10.4 \times 10^6 \text{ psi} & (AL) \\ G = 14.4 \times 10^6 \text{ psi} & (Ti) \end{bmatrix}$

$T_{max} = 368{,}800$ ft.lb.

$T_{Max} = 4425600$ in.lb.

Wt. of Titanium bars : $\rho A l$
$= 49$ lbs.

$\theta_{max} = \frac{(T_{max})(48 \ in)}{(14.4 \times 10^6)\left(\frac{\pi(4)^4}{32}\right)}$

$\theta_{max} = .587 \text{ rad} = 33.6°$

Aluminum: $S_{sy} = .5(73 \text{kpsi}) = 36.5$ kpsi ($\rho = .096 \ ^{lb}/_{in^3}$)

$\sigma = \frac{F}{A}$ $\sigma = \frac{S_{sy}}{n} = \frac{36.5 \text{ kpsi}}{1.5} = 24.3$ kpsi

$\sigma_{max} = 24.3$ kpsi

$F = 24.3 \text{ kpsi} (2\pi \ in^2)$

$F_{max} = 152.9$ kips $T_{Max} = 152{,}900$ ft.lb
$= 1834800$ inlb

Total bar weight $= \rho A l$
$= (48 \ in)(2\pi \ in^2)(.096)$
$\cong \underline{28 \text{ lb}}$

$\theta_{max} = \frac{T_{max} \ l}{G J}$

$= \frac{(1834800 \text{ inlb})(48 \ in)}{(10.4 \times 10^6 \text{ psi})\left(\frac{\pi(4)^4}{32}\right)} = .337$ rad

$= 19.3°$

Alloy #2 (AL) $S_{sy} = .5(47{,}000 \text{psi}) = 23{,}500$ psi

$\sigma_{sy} = \frac{S_{sy}}{n} = 15.67$ kpsi

$F_{MAX} = \sigma_{sy} A = (15.67 \text{ kpsi})(2\pi \ in^2) = 98.45$ kips

$T_{max} = 1181400$ in-lb

$\theta_{max} = \frac{T_{max} \ l}{G J} = .217 \text{ rad} = 12.4°$

$\boxed{L/C = 1.5}$ $\boxed{\beta = 45^\circ}$

$C = 6'$
$L = 9'$
$b = 4.5'$
$b/\delta = \tan 45(2 + \tan^2 45)$
$b/\delta = 3.0$
$\therefore$
$\delta = 1.5$ (Vehicle = 3' high)

$h_{max} = 4.5 \sin 45 - (1.5/\cos 45 - 4.5 \cos 45)$

$h_{max} = 4.24'$ — highest step negotiable.

$C = 7'$
$L = 10.5'$
$b = 5.25'$
$b/\delta = 3.0$
$\delta = 1.75'$ (Vehicle = 3.5' high)
$h_{max} = 4.95'$

$\boxed{L/C = 1.8}$ $\boxed{\beta = 45^\circ}$

$C = 6'$
$L = 10.8'$
$b = 5.4'$
$b/\delta = 3.0$
$\delta = 1.8$ (vehicle = 3.6' high)
$h_{max} = 5.4 \sin 45 - (1.8/\cos 45 - 5.4 \cos 45)$
$h_{max} = 5.09'$

$C = 7'$
$L = 12.6'$
$b = 6.3'$
$\delta = 2.1'$ (Vehicle = 4.2 feet high)

$h_{max} = 6.3 \sin 45 - (2.1/\cos 45 - 6.3 \cos 45)$

$h_{max} = 5.94'$

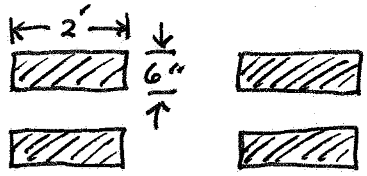
$\boxed{\beta = 30^\circ}$

$C = 6'$
$L = 9.0'$
$b = 4.5'$
$b/\delta = \tan 30(2 + \tan^2 30) = 1.35$
$\delta = 3.33$ (Vehicle = 6.66 ft. high)
$h_{max} = 4.5 \sin 30 - (3.33/\cos 30 - 4.5 \cos$
$h_{max} = 2.202'$

$C = 7'$
$L = 10.5'$
$b = 5.25'$
$b/\delta = 1.35$
$\delta = 3.89'$ (7.8' ht.)
$h_{max} = 5.25 \sin 30 - (3.89/\cos 30 - 5.25 \cos$
$h_{max} = 2.57'$

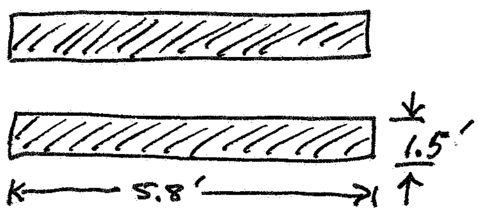
OUR LENGTH = 10 ft.
OUR WIDTH = 6 ft
$\frac{L}{C} = 1.67$
$b = 5$ ft.
$\frac{b}{\delta} = \tan\beta\,(2 + \tan^2\beta)$
Let $\delta = 3.25/2$ (design height)
$\therefore$
$\frac{5}{3.25/2} = \tan\beta\,(2 + \tan^2\beta)$
$\therefore \beta = 45^\circ$
obstacles that create a 45° angle between the vehicle and ground can be traversed.

$$\text{Nominal ground pressure} = \frac{\text{Vehicle wt.}}{\text{area of wheels in ground contact}}$$



wheels → 2' wide contact
(6" wide tread)

Area = 4 (2' x .5') = 4 ft^2



actual tread contact with ground (not final)

Tracks → 1' wide, 5.8' Long
Area = 2 (5.8' x 1.5') = [redacted] ft^2 17.4

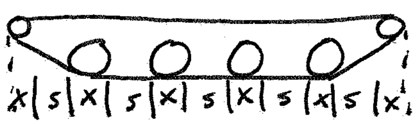
$$\underset{(\text{wheels})}{NGP} = \frac{\text{Vehicle wt.}}{4}$$

$$\underset{(\text{track})}{NGP} = \frac{\text{VEHICLE}}{17.[redacted]4}$$

– clearly NGP will be higher for wheels.

Contact Area [redacted] 17.4 to 4;
[redacted] [redacted] greater than 4:1
(for 1' wide tread → ratio is almost 3:1)

Actual wheel spacing for Treads: (preliminary)



let x = wheel width
let s = space width
∴ $\boxed{6x + 5s = \text{length}}$

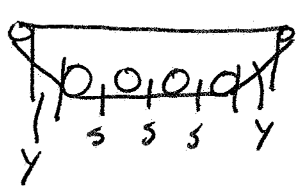
[redacted] for length (total) of 9.0 ft.

x	s (feet)
.8	.84
.9	.72
1.0	.6
1.1	.48
1.2	.36

length = 10.5 ft.
(other extreme from previous calculations).

x	s (feet)
.9	1.02
1.0	.9
1.1	.78
1.2	.66

(#2) Actual Spacing: ([redacted])



x = wheel spacing

idler & sprocket = 2/3 size of Road wheel

$2y + 3s + 4x + \frac{2}{3}x = 9.0$ $\boxed{x = 1 \text{ ft}}$

$2y + 3s = 4.33$

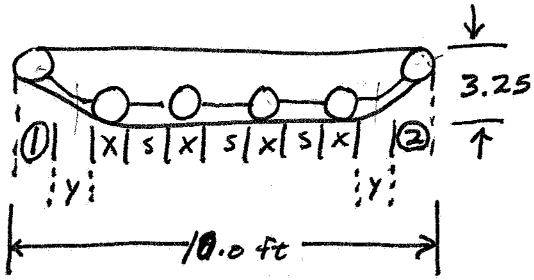
Let $y = \frac{4}{3}s$

$\therefore \frac{17}{3}s = 4.33$

$s = .76'$ or $9''$

$\therefore y = 1$ ft.

Spacing: (#3)



① = 12 in idler wheel
② = 12 in sprocket
x = ~~20 in wheels~~ 18 in wheels
s = Road wheel spacing
y = more spacing

$4(\cancel{20}\,18'') + 3(s) + 2(y) + (24'') = 10.0'$

$3s + 2y = 2.0'$

$\frac{3}{2}y + 2y = 2$

$3\frac{1}{2}y = 2$

$y = .57'$ or 7 in

$s = 3.5$ ft.

Let $s = \frac{1}{2}y$
or
Let $s = y$

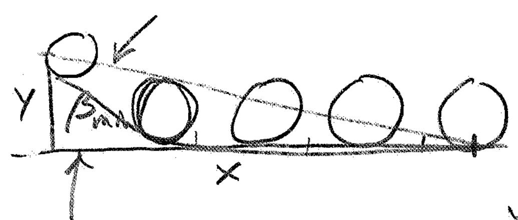
$5y = 2.0'$

$\boxed{s = y = 4.8''}$

Finally,

x = 18 inches (wheels)
idler (front) wheel = 12 inches
sprocket (drive) wheel = 12 inches
total length = 10 ft.

For smaller vehicle → lose one wheel.



$\beta_{min} = \tan^{-1}\left(\frac{Y}{X}\right)$

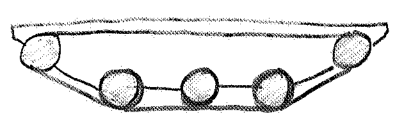
$Y = 33$ in $(39'' - 6'')$

$X = 3(18'') + 1(9'') + 4(4.8'') + 1(12'')$

$X = 7.85'$

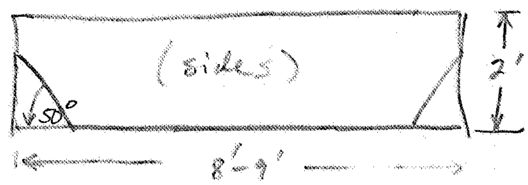
30" + 6"

$\beta_{min} = \tan^{-1}\left(\frac{33\text{ in}}{94.2\text{ in}}\right) = 19.31°$



Hull thickness = $\frac{1}{16}'' - \frac{3}{16}''$ Aluminum

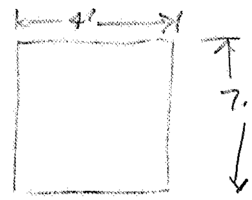
Hull Area =



$\frac{1}{2}bh = \frac{1}{2}(2')(2.4')$

$\sim 18\ ft^2 \times 2 = 36\ ft^2$

Bottom ⇒



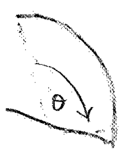
$= 30\ ft^2$

BACK ⇒ $2.5' \times 4'$ plate = $10\ ft^2$
Front ⇒ $1' \times 4'$ plate = $4\ ft^2$
TOTAL = $80\ ft^2$

$\rho = .096\ lb/in^3$

$1/16''$ plate thickness → Volume = $.4167\ ft^3 = 720\ in^3$ ⇒ 69 lbs.
$3/16''$ plate thickness → Volume = $1.25\ ft^3 = 2160\ in^3$ ⇒ 207.4 lbs.
$1/8''$ plate thickness → Volume = $.8334\ ft^3 = 1440\ in^3$ ⇒ 138.24 lbs.

Bin Area ⇒



$\theta \cong 100°$

$s = r\theta = (2') (1.745\ rad)$
$s = 3.5\ ft$
width ≅ 5 ft
total Area = $17.5\ ft^2$
total volume = $(17.5\ ft^2)(1/16'') = .0911\ ft^3$
= 15.12 lbs.

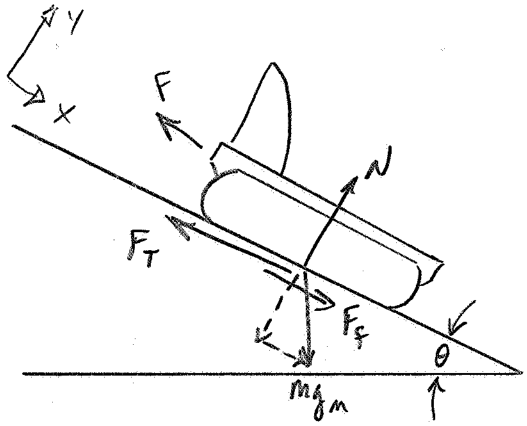
Pump weight = 54 lbs.

Torsion bars = 112 lbs.
wheels = 74 + 120 = 194 lbs.
Motor + ~~transmission~~ = 200 lbs.

Total weight (excluding tiller assembly) = 782 lbs ($\approx$ 354 kg)

Tiller estimate : ~100 lbs.

Total = $\boxed{882\ lbs}$



F_T = Traction force
F_f = Frictional force
$F = ma$
$g_m = 1.635\ m/sec^2$

$\mu = .625$ (regolith)
$m = 400$ kg (approximate Vehicle mass)

Assume 20% slip (conservative)

$$\Sigma F_x = -F + \mu m g \cos\theta + mg\sin\theta + F_T = \cancel{ma}^{\,0} \quad \text{(constant Velocity)}$$

$$F = \mu m g \cos\theta + mg\sin\theta + 34.1\ N \quad \text{(see Traction force calculation)}$$

$$F = (.625)(400\,kg)(1.635\ m/sec^2)\cos\theta + (400\,kg)(1.635\ m/sec^2)\sin\theta + 34.1\ N$$

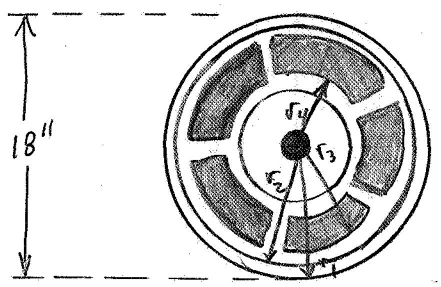
$$F = 408.75\cos\theta + 654\sin\theta + 34.1\ N$$

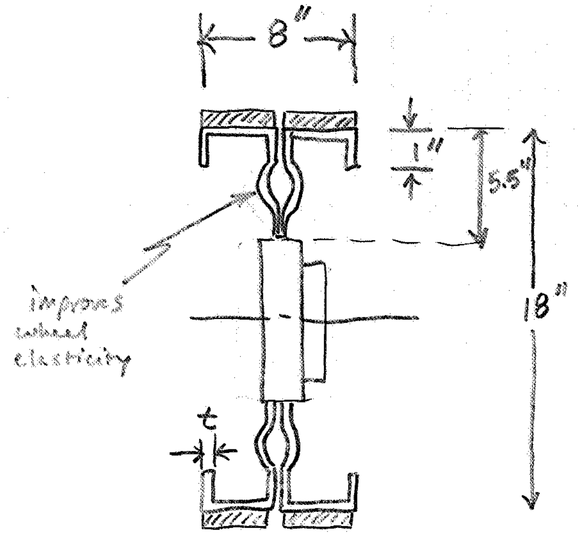
FORCE (N)	Power (W)	ANGLE (deg)	Power (hp)
498.3	895.94	5	1.201
524.7	943.41	7.5	1.265
550.2	989.26	10.0	1.327
574.7	1033.31	12.5	1.386
598.2	1075.56	15.0	1.442
620.6	1115.84	17.5	1.496
641.9	1154.14	20.0	1.548
662.0	1190.28	22.5	1.596
680.9	1224.26	25.0	1.642
698.6	1256.1	27.5	1.684
715.1	1285.75	30.0	1.724

Power = $F \cdot V$ $\quad V = 1.798\ m/sec$ (or 4 mi/hr)

Using the maximum Power and working through the Drive shaft torque equations, a maximum motor power of 3.76 hp is required, For a slope of 30°.

ROAD WHEELS





$r_1 = 9$ in
$r_2 = 8$ in
$\pi(r_1^2 - r_2^2) = \pi(9^2 - 8^2) = 17\pi$
$17\pi t$

$t = .25''$

$2\pi(9) = 18\pi(8) = 144\pi$

$r_3 = 7$ in
$r_4 = 4$ in
space Area $= \pi(r_3^2 - r_4^2) - 2''(r_3 - r_4)$
$= \pi(49 - 16) - 2''(3 \text{ in})$
$= (33\pi - 6)\text{ in}^2$
$\cong 98 \text{ in}^2$
Metal area $= \pi(8)^2 - 98 \text{ in}^2$
$= 103 \text{ in}^2$

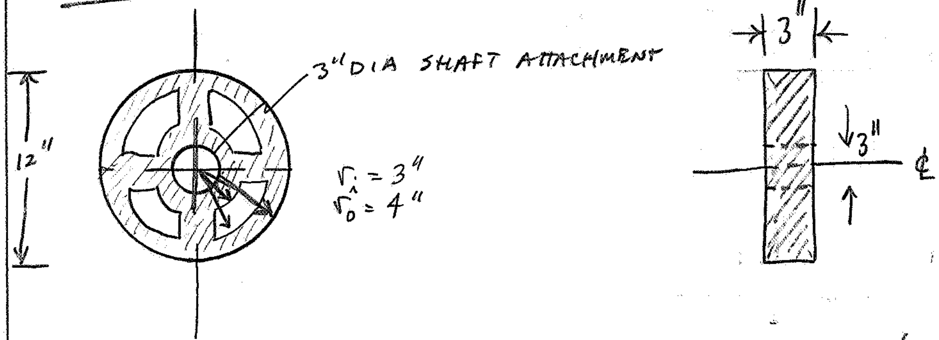
Total metal volume $\left.\right\} \rightarrow 17\pi(t) + 144\pi(t) + 103(t) =$
$= 609t \text{ in}^3 = 152 \text{ in}^3$

mass $= (.096 \text{ lbm/in}^3)(152 \text{ in}^3)$
$= 14.6$ lbm
$\cong 15$ lbm

moon weight $= 2.5$ lb

ROAD, IDLER & SPROCKET DESIGN: (NOT to SCALE)

IDLER:

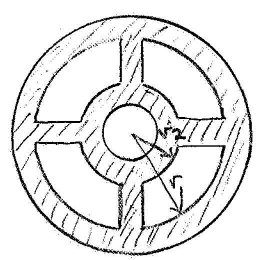


Approximate MASS → $(.096\ \text{lb/in}^3)(193\ \text{in}^3) = 18.5\ \text{lbm}$

Sprocket wheel will be same size only will drive the tread.

		moon weight
TOTAL MASS OF 2 idler wheels, 2 sprockets	~74 lbm	(12.33 lb)
4 idler wheels, 2 sprockets	~111 lbm	(18.5 lb)

$S = r\theta$



$r_1 = 5\ \text{in}$
$r_2 = 2\tfrac{1}{2}\ \text{in}$

Total Area $= \pi(6)^2 = 36\pi$
ring Area $= \pi\left((5)^2 - (2.5)^2\right) = 18.75\pi$
$\sim 4(2\tfrac{1}{2}" \times 1") = 10\ \text{in}^2$

$18.75\pi\ \text{in}^2 - 10\ \text{in}^2 \cong 49\ \text{in}^2$

Total metal Area $= (36\pi - 49)\ \text{in}^2$
$\cong 64\ \text{in}^2$

$\therefore$ Total Volume $= 193\ \text{in}^3$

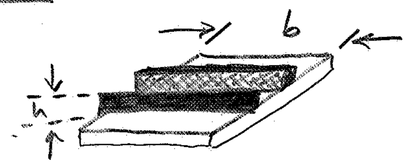
Let slip vary : i = 1, 5, 10, 15, 20, 25, 30 percent

V_V = vehicle speed

V_W = peripheral speed of track $= \frac{V_V}{1-i}$

Tractive force ⇒ $F = \frac{b h \gamma i V_W^2}{g}$

where b = track width
h = tread height
$\gamma = 120 \text{ lb/ft}^3$



For our vehicle : $b = 1'$
$h = 1/2'' = .04167 \text{ ft}$

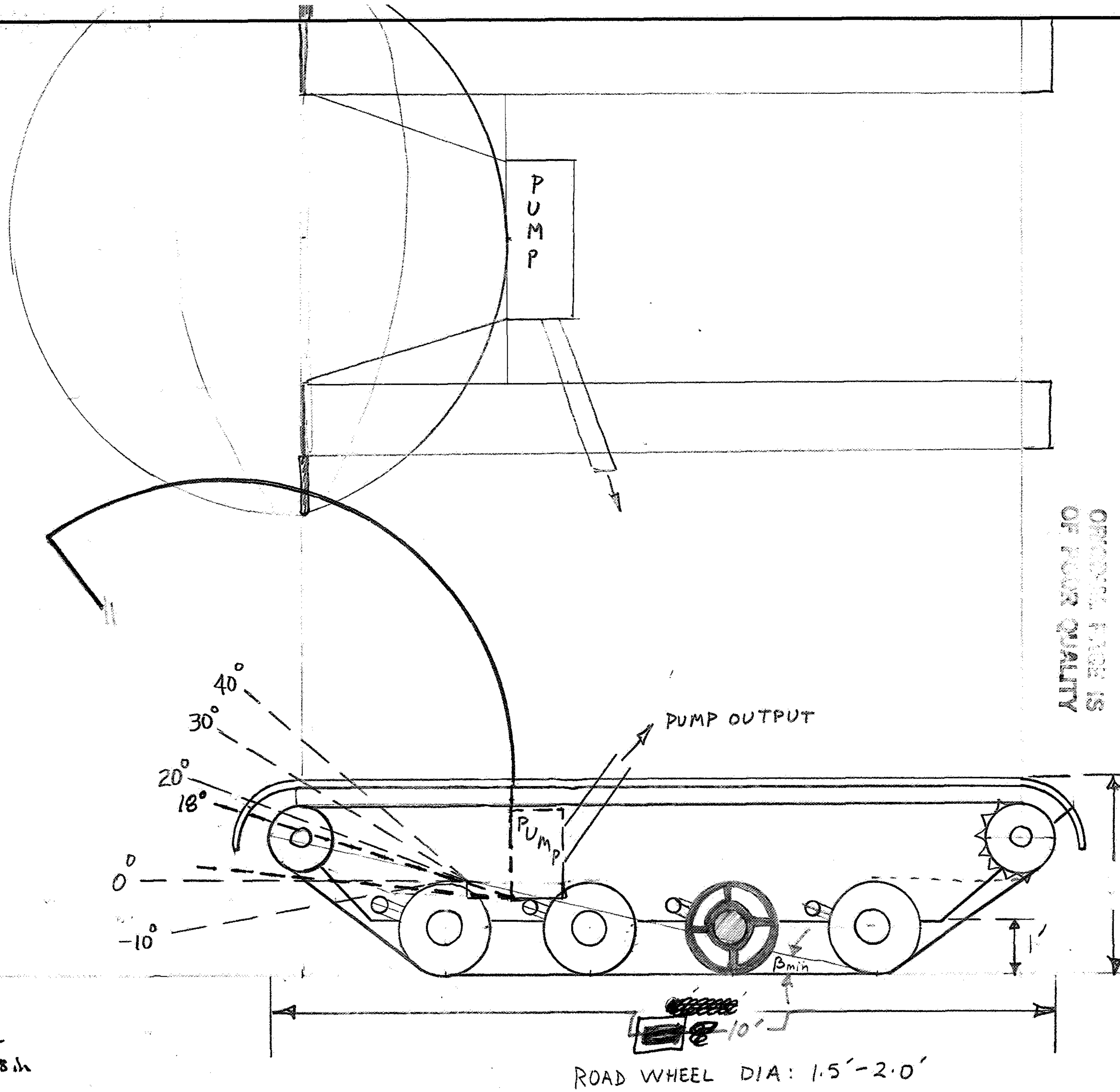
$V_W = \frac{V_V}{1-i}$ operating Velocity $\simeq$ 3.5 mi/hr = 5.13 ft/sec

$$F_{traction} = \frac{(1\text{ ft})(.04167\text{ ft})(120\text{ lb/ft}^3)(i)\left(\frac{5.13}{1-i}\frac{\text{ft}}{\text{sec}}\right)^2}{g = \frac{32.2\text{ ft/sec}^2}{6} = 5.37\text{ ft/sec}^2}$$

$$F_{traction} = 24.5\left(\frac{i}{(1-i)^2}\right)\ \text{lb}_f \quad \text{or}$$

$$F_{traction} = 109.025\left(\frac{i}{(1-i)^2}\right)\ N$$

i (%)	$F_{traction}$ (N)
1	1.11
5	6.04
10	13.46
15	22.63
20	34.10
25	48.50
30	66.80
35	90.32
40	121.14
45	162.20
50	218.1



$$\beta_{min} \cong \tan^{-1}\left(\frac{2.75}{7.85}\right)$$

$$\beta_{min} = 19.31^{\circ}$$

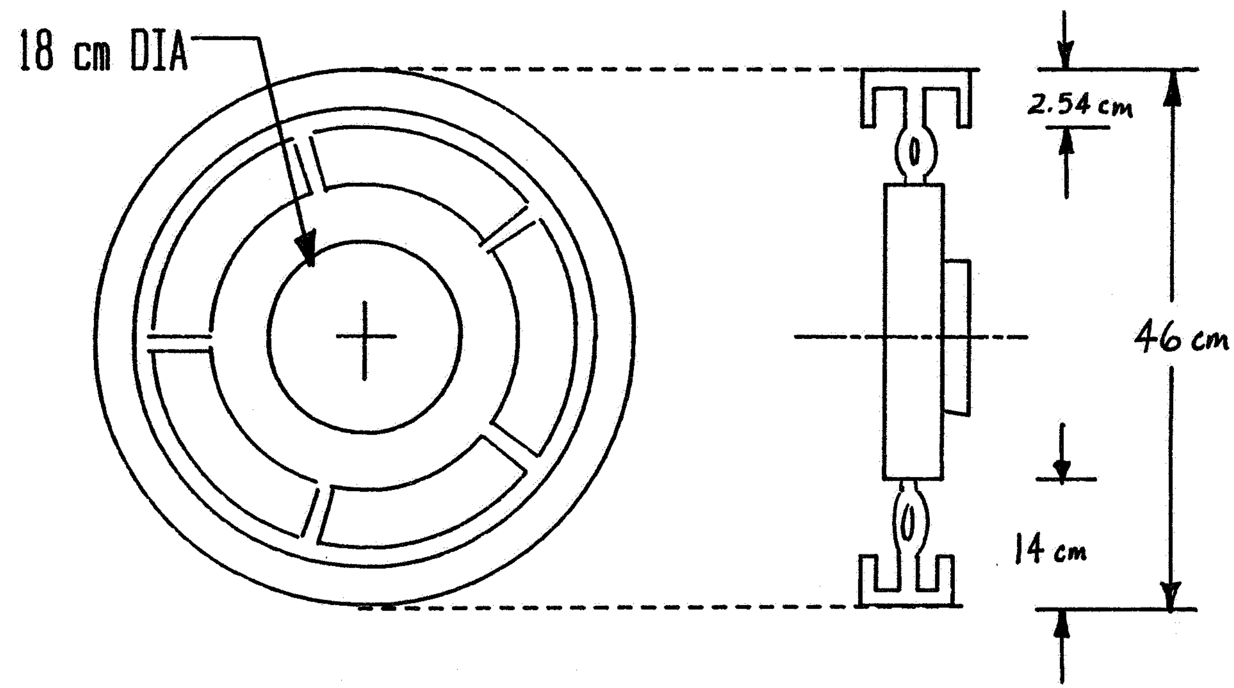
ROAD WHEEL DIA = 18 in
IDLER = SPROCKET = 12 in
Road wheel Spacing = 4.8 in

DRAWN BY

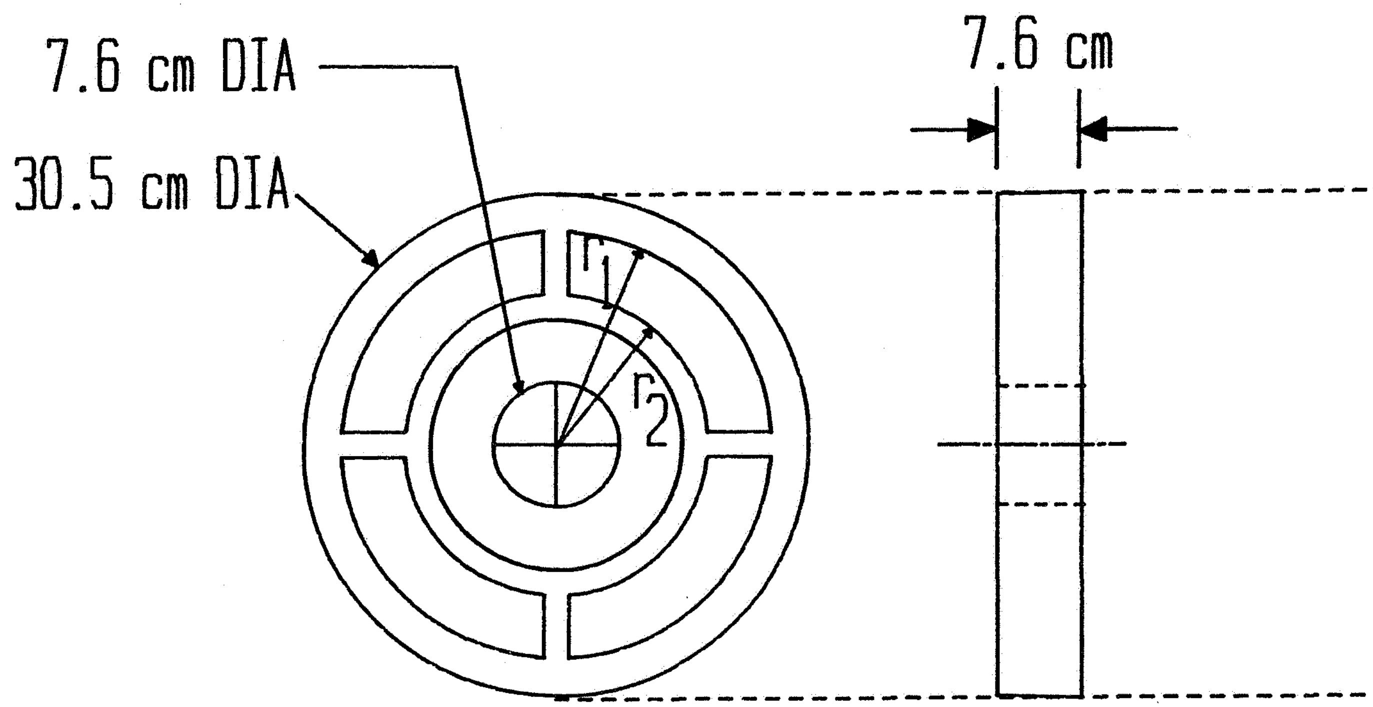
FILE NO.

DRAWING

WHEEL DESIGN: ROAD WHEELS



WHEEL DESIGN: IDLER AND DRIVE SPROCKET



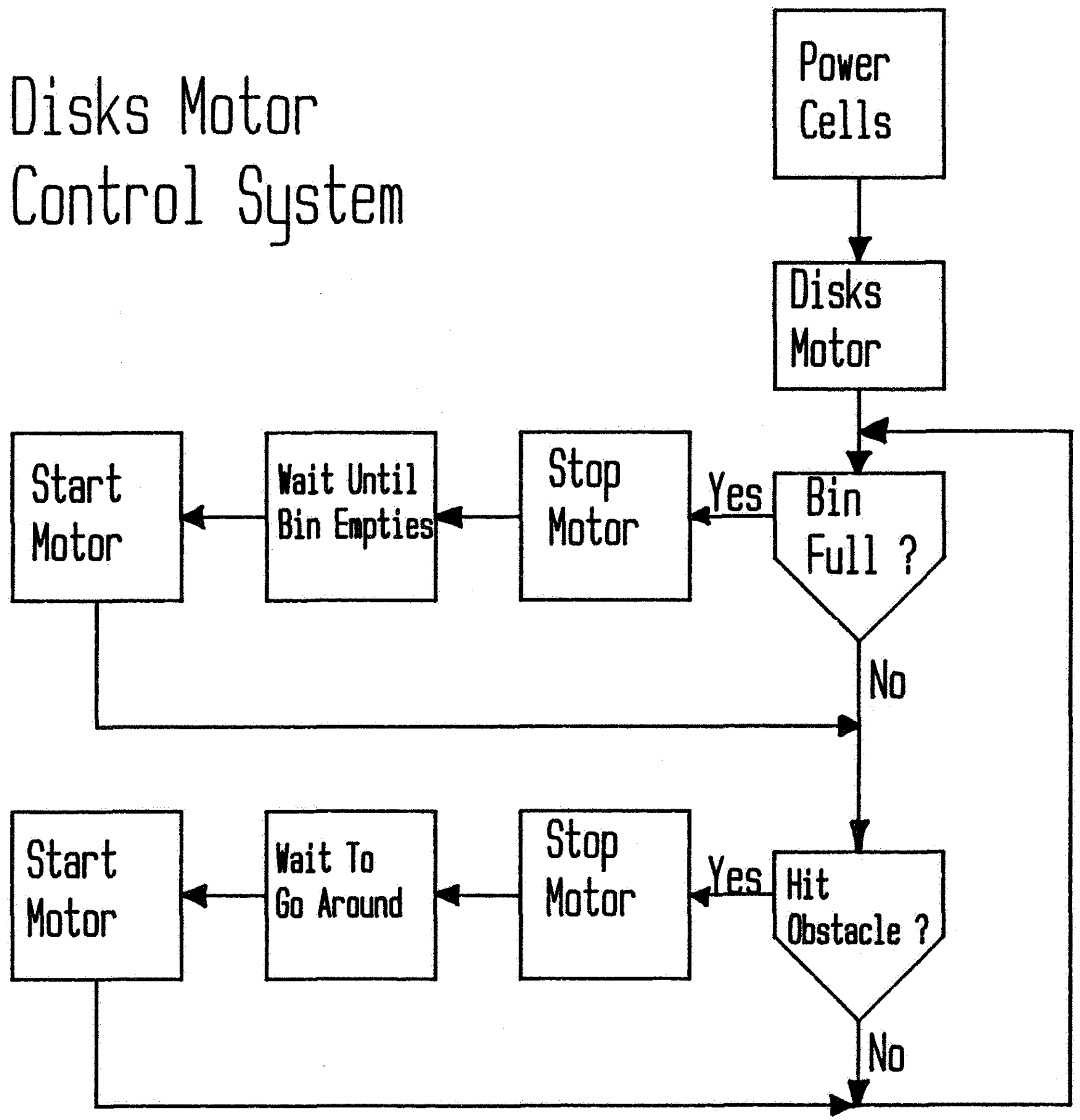
$r_1 = 13$ cm

$r_2 = 6.5$ cm

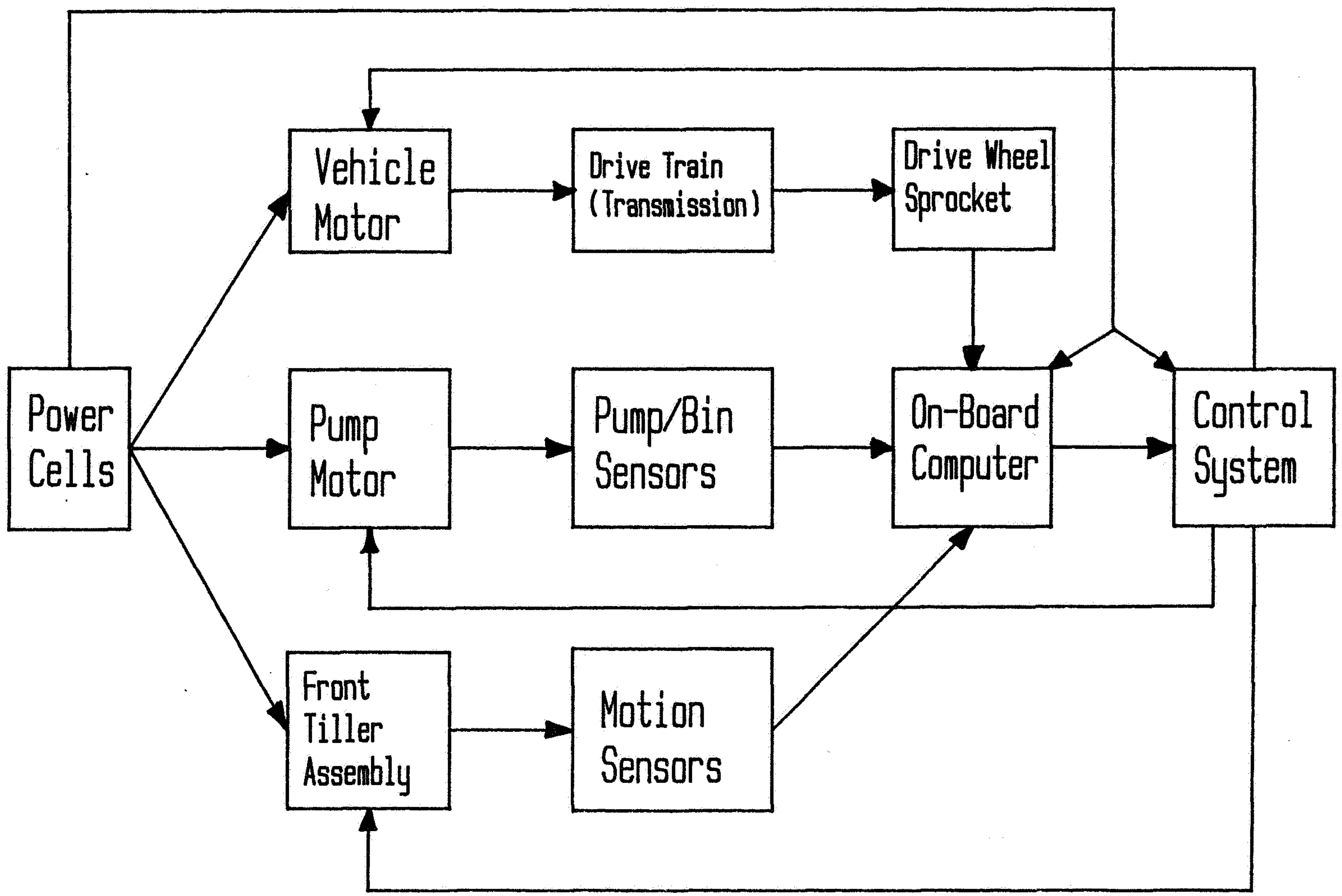
Appendix 2

Control Schematics

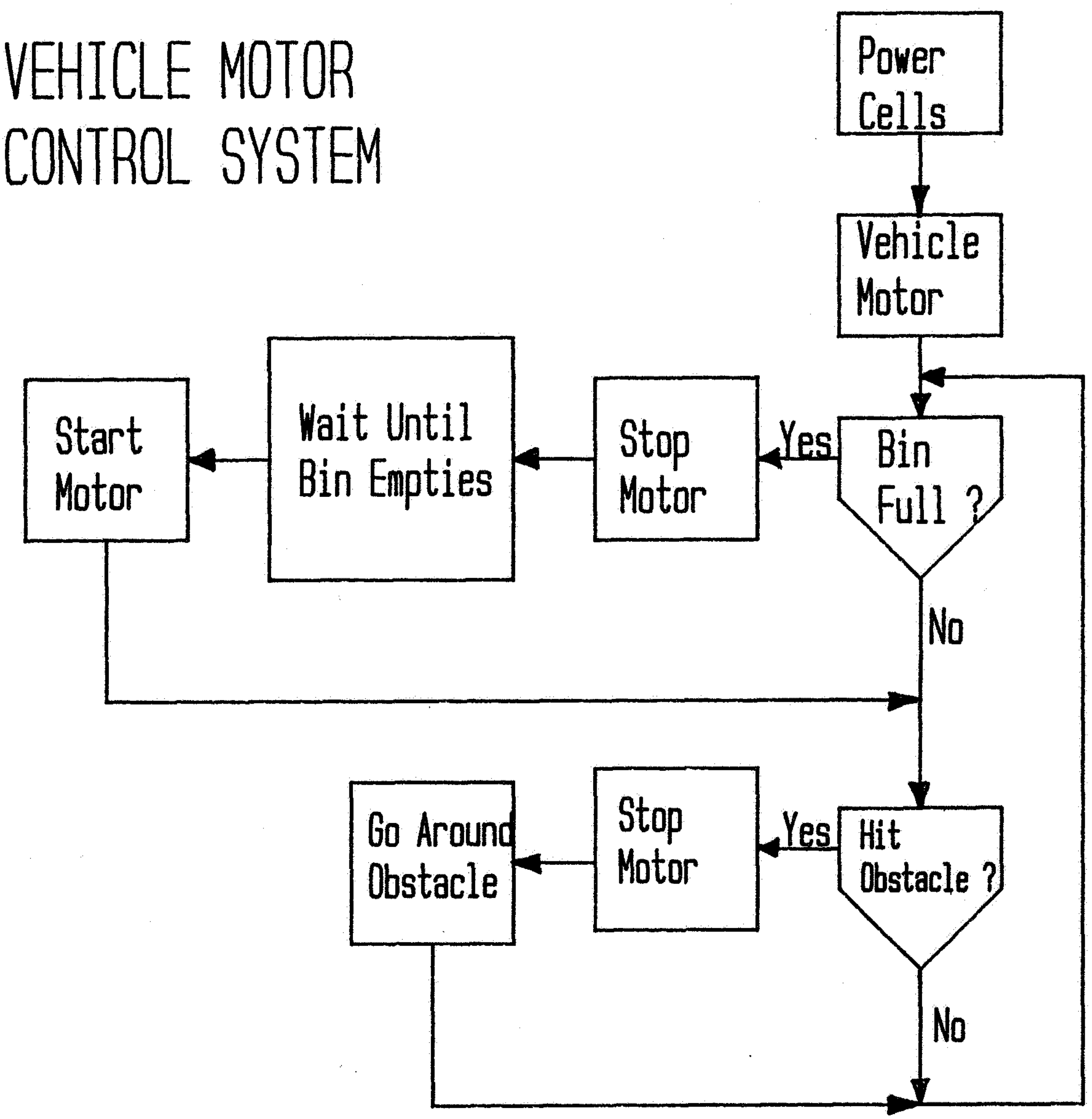
Disks Motor Control System



ELECTRICAL AND CONTROLS SYSTEMS



VEHICLE MOTOR CONTROL SYSTEM

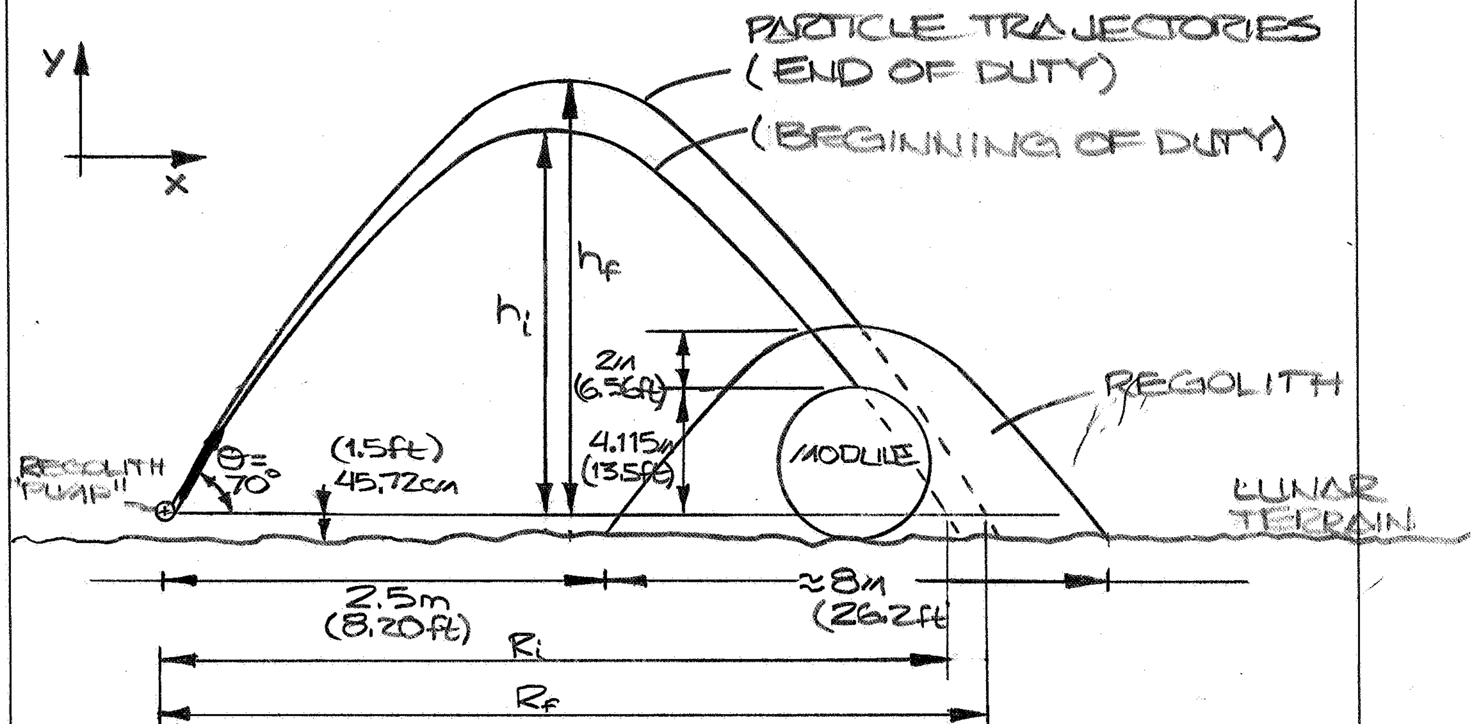


Appendix 3

Pump

IMPELLER DESIGN

PERFORMANCE PARAMETERS



CALCULATIONS

GIVEN: $\theta = 70°$; $g = 1.635 \frac{m}{s^2}$

$$R_i = \frac{V_{0,i}^2 \sin(2\theta)}{g} = \frac{V_{0,i}^2 \sin(140)}{1.635} = .3931\, V_{0,i}^2$$

$$h_i = \frac{V_{0,i}^2 \sin^2\theta}{2g} = \frac{V_{0,i}^2 \sin^2(70)}{2(1.635)} = .2700\, V_{0,i}^2$$

PARTICLE PATH EQUATION:

$$y_i = \tan\theta \cdot x_i - \frac{g}{2V_{0,i}^2\cos^2\theta}x_i^2 = \tan(70)x_i - \frac{1.635}{2(V_{0,i}^2\cos^2(70))}x_i^2$$

FROM THE FIGURE, $y_i = 4.115\,m$ AND $x_i = 6.5\,m$

$$y_i = 2.747x_i - \frac{6.988}{V_{0,i}^2}x_i^2 \Rightarrow 4.115 = 2.747(6.5) - \frac{6.988(6.5)^2}{V_{0,i}^2}$$

SOLVING FOR $V_{0,i}^2$,

$$V_{0,i}^2 = \frac{6.988(6.5)^2}{4.115 - 2.747(6.5)} = 21.487 \Rightarrow V_{0,i} = \boxed{4.635 \frac{m}{s}}$$

$$\therefore\; R_i = .3931(4.635)^2 = \boxed{8.445\,m}$$

$$h_i = .2700(4.635)^2 = \boxed{5.800\,m}$$

FOR FINAL CONDITIONS,

$Y_f = 6.115\,m$ AND $X_f = 6.5\,m$

SIMILAR CALCULATIONS YIELD

$V_{0,f} = 5.015\,\frac{m}{s}$

$R_f = 9.885\,m$

$h_f = 6.789\,m$

THEREFORE IMPELLER SPEED MUST VARY TO ACCOMMODATE PARTICLE VELOCITES OF $4.635\,\frac{m}{s}$ TO $5.015\,\frac{m}{s}$.

IMPELLER DESIGN CALCULATIONS

CALCULATIONS ARE BASED ON THE FOLLOWING:

*	n = # OF IMPELLER BLADES	= 6
*	R_o = OUTSIDE IMPELLER BLADE RADIUS	= .1016 m (4.0 IN)
*	R_I = INSIDE IMPELLER BLADE RADIUS	= .0381 m (1.5 IN)
*	t = IMPELLER BLADE THICKNESS	= .0127 m (0.5 IN)
*	W = IMPELLER BLADE WIDTH	= .1016 m (4.0 IN)
	ρ_R = REGOLITH DENSITY	= 1750 KG/m^3
	V = EXIT REGOLITH VELOCITY	= 5.015 m/s
	ρ_I = IMPELLER MATERIAL DENSITY (TITANIUM)	= 4505 $\frac{kg}{m^3}$

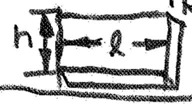
* DIMENSION CHOOSEN TO OPTIMIZE ENERGY VS. WEIGHT EFFICIENCY

CALCULATION OF IMPELLER WEIGHT

V_1

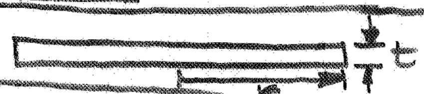
$$V_1 = \pi(r_o^2 - r_i^2)(t) = \pi((.1016)^2 - (.0381)^2)(.00635) = .000176967\,m^3 = 176.967\,cm^3$$

V_2



$$V_2 = ndlh = 6(.0127)(.0699)(.1016) = .00054\,m^3 = 541.16\,cm^3$$

V_3



$$V_3 = \pi r^2 t = \pi(.1016)^2(.00635) = .000205\,m^3 = 205.925\,cm^3$$

V_4

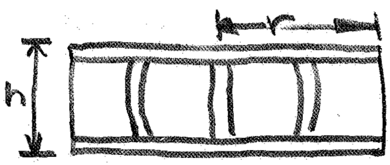
$$V_4 = \pi\left(\frac{d_o^2 - d_i^2}{4}\right)h = \frac{\pi}{4}((.0254)^2 - (.0127)^2)(.0254) = .000009652\,m^3 = 9.652\,cm^3$$

∴ TOTAL VOLUME OF IMPELLER $= V_1 + V_2 + V_3 + V_4$

$$V_{IMP,TOTAL} = 176.967 + 541.16 + 205.925 + 9.652 = 933.704\,cm^3 = .000933704\,m^3$$

$$MASS = \rho_I V_I = (4505\,kg/m^3)(.000933704\,m^3) = 4.2063\,kg$$

VOLUME OF REGOLITH / REVOLUTION; V_R



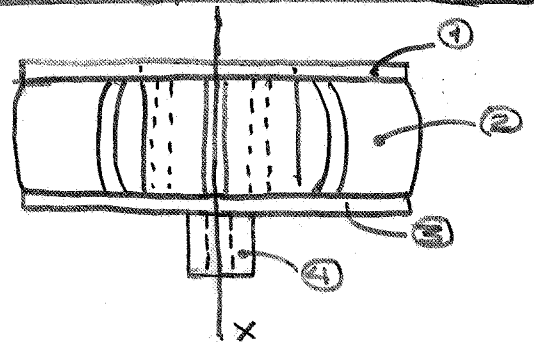
$$V_R = V_{CYL} - V_{IMP,TOT}$$
$$= \pi r^2 h - 933.704\,cm^3$$
$$= \pi(10.16\,cm)^2(11.43\,cm) - 933.704\,cm^3$$
$$= 2772.962\,cm^3 = .00277\,m^3$$

$$\therefore MASS = \rho_R V_R = (1750\,kg/m^3)(.00277\,m^3) = 4.8475\,kg$$

THEREFORE,

$$WEIGHT_{TOTAL} \approx 4.2063\,kg + 4.8475\,kg = 9.0538\,kg$$

MOMENT OF INERTIA (IMPELLER), $I_{x,I}$



$$I_{x,1} = \frac{m_1(r_o^2+r_i^2)}{2} = \frac{\rho_I V_1(r_o^2+r_i^2)}{2} = \frac{(4505\frac{kg}{m^3})(.000176967m^3)[(.1016)^2+(.0381)^2]}{2}$$

$$= .004693 \text{ kg m}^2$$

$$I_{x,2} = n(I_c + m_2 d^2) = n\left[\frac{m_2(t^2+(r_o-r_i)^2)}{12} + m_2 d^2\right]$$

$$= n\left[\rho_I V_2 \frac{[t^2+(r_o-r_i)^2+d^2]}{12}\right] = 6(4505\frac{kg}{m^3})(.000554)\left[\frac{(.0127)^2+(.0699)^2}{12}+(.108)^2\right]$$

$$= .17639 \text{ kg m}^2$$

$$I_{x,3} = \frac{m_3 r_o^2}{2} = \frac{\rho_I V_3 r_o^2}{2} = \frac{(4505)(.000205)(.1016)^2}{2} = .004767 \text{ kg m}^2$$

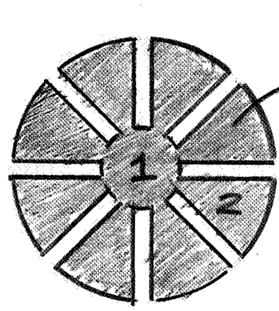
$$I_{x,4} = \frac{m_4(r_o^2+r_i^2)}{2} = \frac{\rho_I V_4((.0127)^2+(.00635)^2)}{2} = (4505)(.00000952)\times(.000100806)$$

$$= 4.38328\times10^{-6} \text{ kg m}^2$$

$$\therefore I_{x,I} = I_{x,1}+I_{x,2}+I_{x,3}+I_{x,4} = .004693+.17639+.004767+4.38328\times10^{-6}$$

$$= .1858543 \text{ kg m}^2$$

MOMENT OF INERTIA (REGOLITH), $I_{x,R}$



d = DEPTH

$$I_1 = \rho \int r^2 dV \qquad dV = 2\pi r d\, dr$$

$$= \rho \int_0^{R_i} r^2 \, 2\pi r d \, dr$$

$$= 2\pi d \rho \int_0^{R_i} r^3 dr$$

$$= \frac{2\pi d\rho \, r^4}{4}\Big|_0^{R_i} = \frac{\pi d \rho R_i^4}{2} = \frac{\pi(.1016)(1750)(.0381)^4}{4} = 2.943E \text{ KG}\cdot\text{m}^2$$

$$I_2 = \rho \int r^2 dV \qquad dV = d \; dA \Rightarrow dA = r dr d\theta$$

$$= \rho \int_0^{\pi/4} \int_{R_i}^{R_o} d r^3 dr d\theta \qquad dV = d\, r dr d\theta$$

$$= \rho d \int_0^{\pi/4} \frac{r^4}{4}\Big|_{R_i}^{R_o} d\theta$$

$$= \rho d \int_0^{\pi/4} \frac{R_o^4 - R_i^4}{4} d\theta$$

$$= \rho d \, \frac{\pi (R_o^4 - R_i^4)}{16} = \frac{\pi (1750)(.1016)}{16}\left((.1016)^4 - (.0381)^4\right) = .003656$$

$$\therefore I_{x,R,TOT} = I_1 + nI_2 = 2.943E{-}4 + 6(3.646E{-}3) = .02217 \text{ KG}\cdot\text{m}^2$$

TOTAL MOMENT OF INERTIA

$$I_{TOTAL} = I_{x,I} + I_{x,R} = .18585 + .02217 = .20802 \text{ KG}\cdot\text{m}^2$$

TOTAL MECHANICAL POWER NEEDED TO DRIVE IMPELL[ER]

ASSUME t SECONDS TOTAL TIME TO REACH $V = 5.05\frac{m}{s}$

t (SECONDS)	α (RAD/s^2)	ω (RAD/S)	T (N·m)	P (WATTS)	S (m^3/HR)
1	68.65	68.65	14.28	980	684.6
2	34.33	68.65	7.14	490	684.6
3	22.88	68.65	4.76	326	684.6
4	17.16	68.65	3.57	245	684.6
5	13.73	68.65	2.86	196	684.6
6	11.44	68.65	2.38	163	684.6
7	9.81	68.65	2.04	140	684.6
8	8.58	68.65	1.79	122	684.6
9	7.63	68.65	1.59	108	684.6
10	6.87	68.65	1.43	98	684.6

CALCULATIONS ABOVE ARE BASED ON THE FOLLOWING DATA AND EQUATIONS.

DATA

$V = 5.05\frac{m}{s}$; $r_c = .07355m$; $I = .20802\ kg \cdot m^2$; $VOLUME = .00277 m^3$

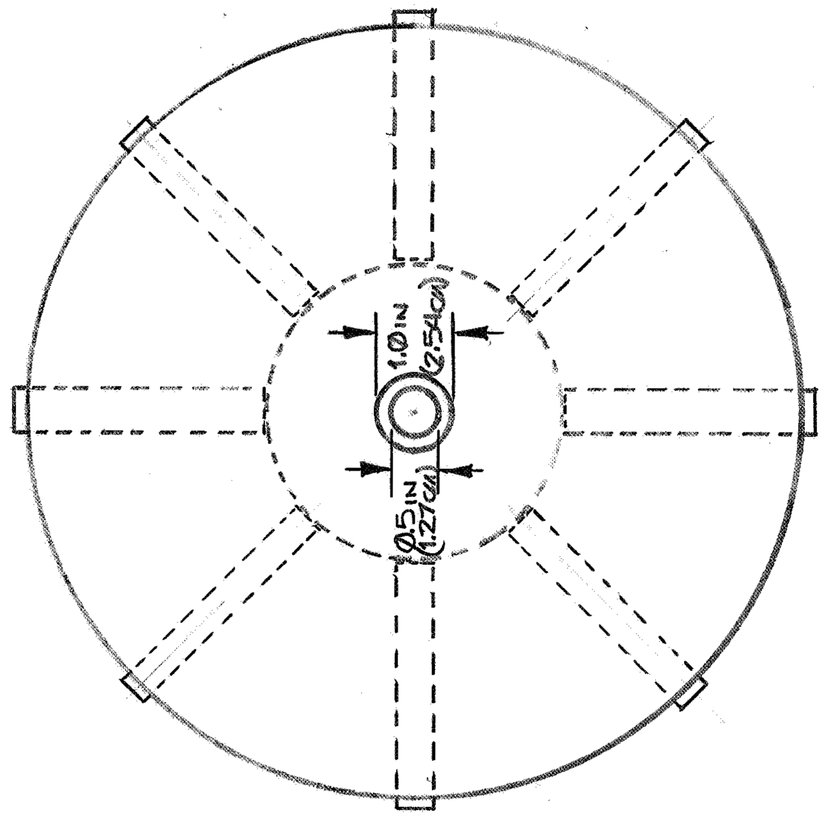
EQUATIONS

$a = \frac{V}{t} \Rightarrow \alpha = \frac{a}{r_c} \Rightarrow T = I\alpha \Rightarrow P = T\omega$

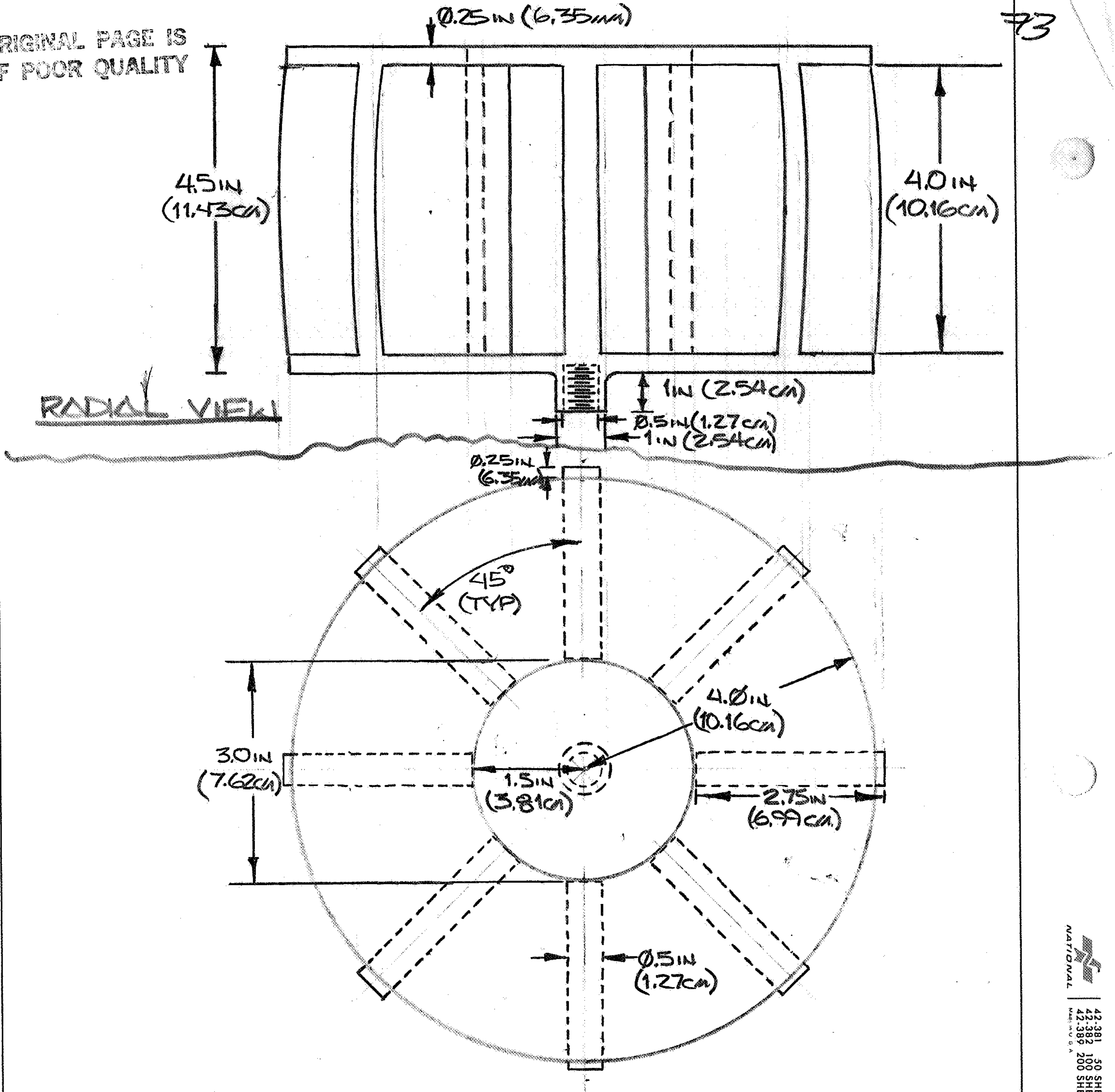
$\omega = \frac{V}{r_c} \Rightarrow S = VOL \times \omega$

THERE THE MOTOR NEEDS TO SUPPLY AT LEAST 14.28 N·m OF TORQUE

NOTE: THE ABOVE CALCULATIONS DISREGARD FRICTIONAL DRAG BETWEEN THE REGOLITH AND IMPELLER CASING SINCE THE INSIDE OF CASING SHALL BE AN EXTREMELY SMOOTH AND HARD SURFACE.

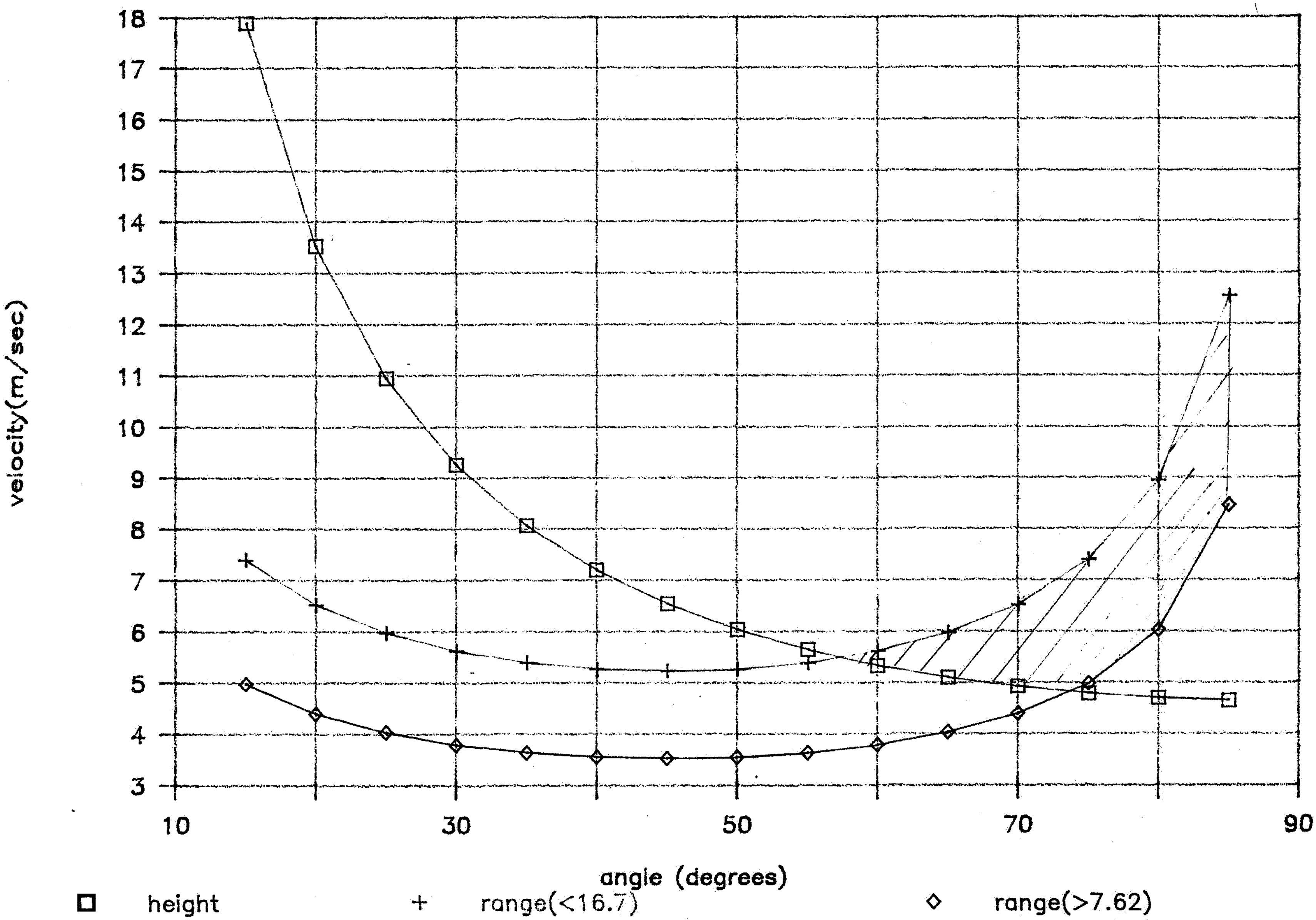


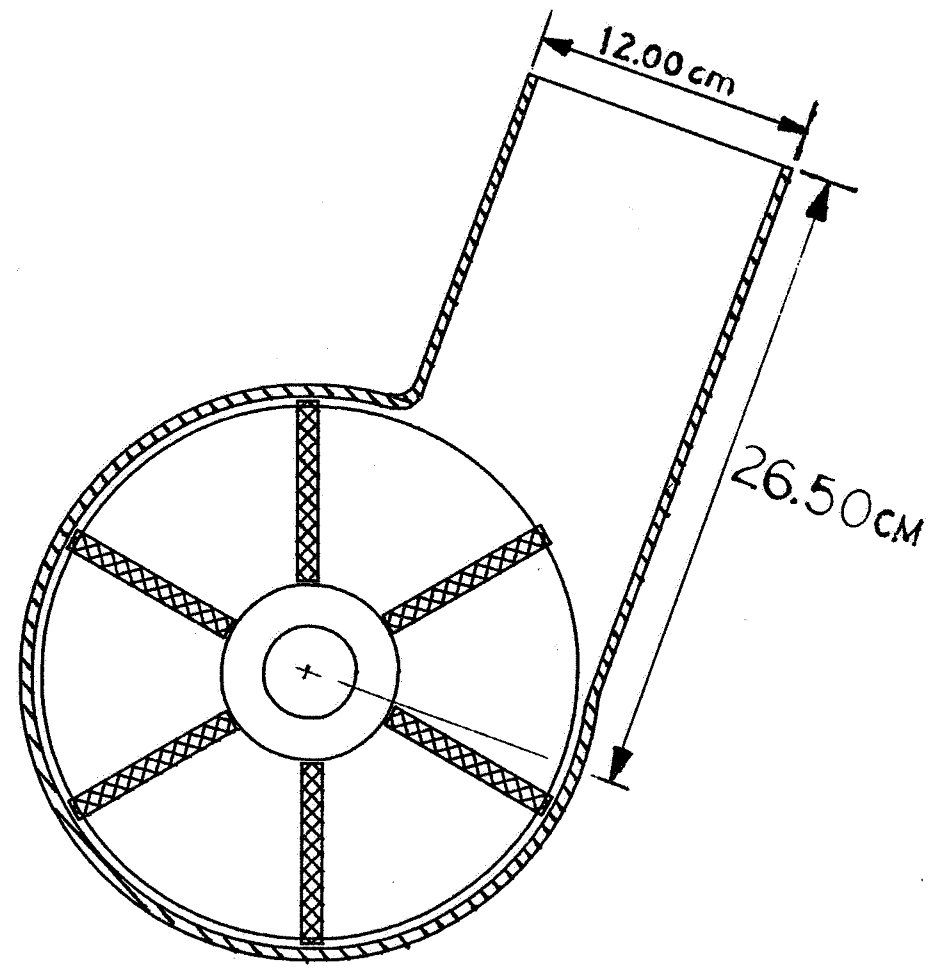
IMPELLER
(MOTOR SIDE)



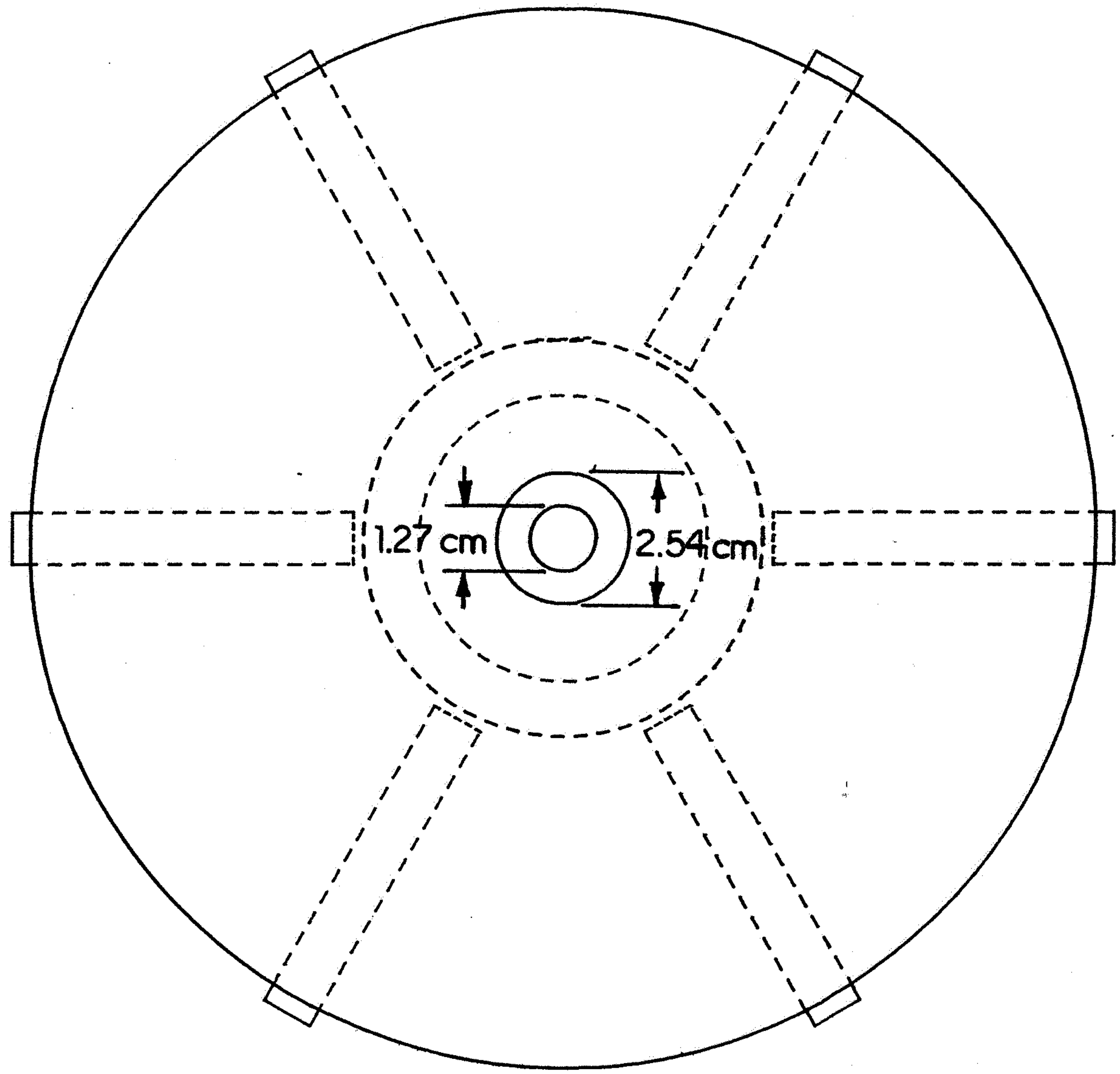
IMPELLER

(SUPPLY SIDE)

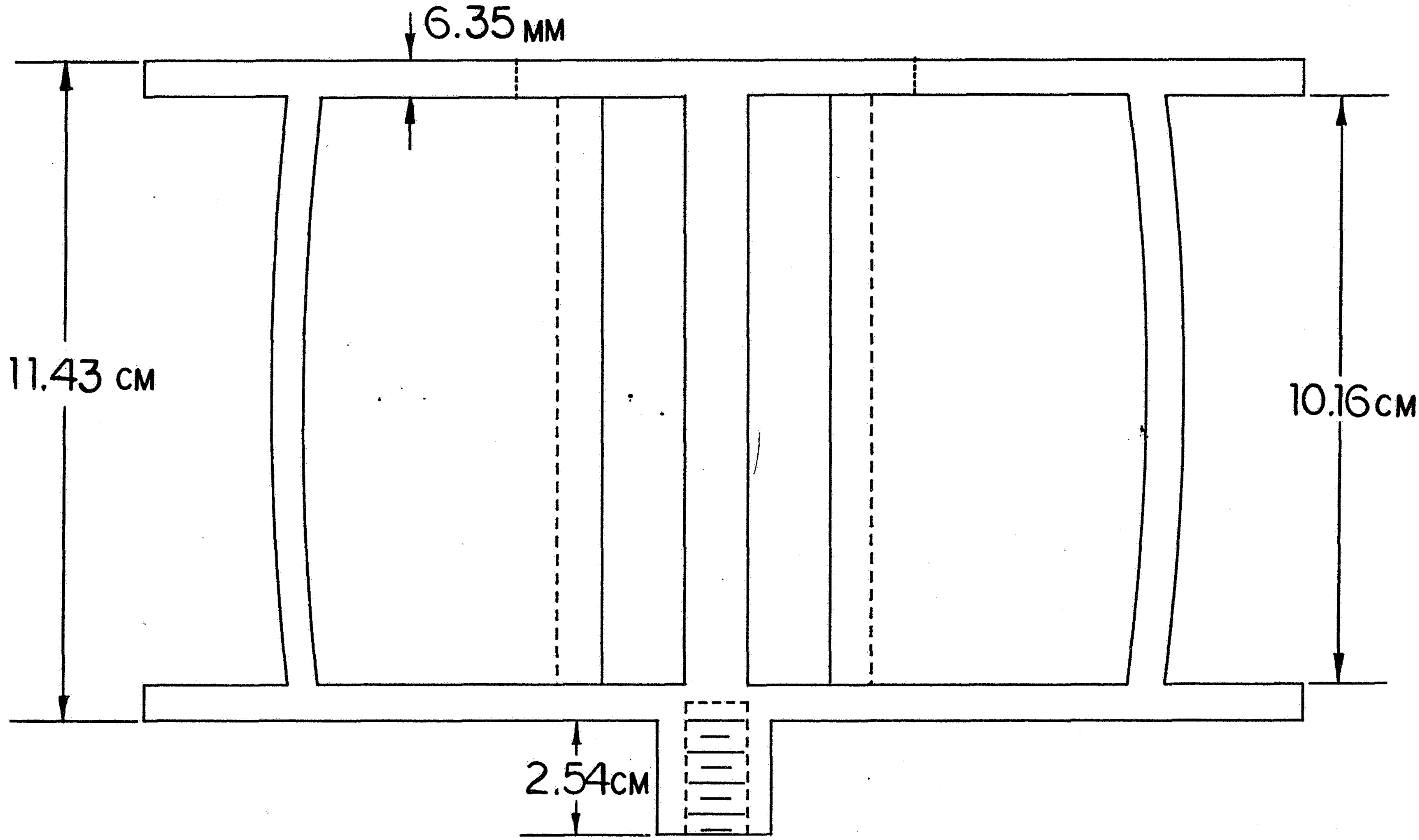

velocity(m/sec)
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
10
30
50
70
90
angle (degrees)
height
+ range(<16.7)
range(>7.62)



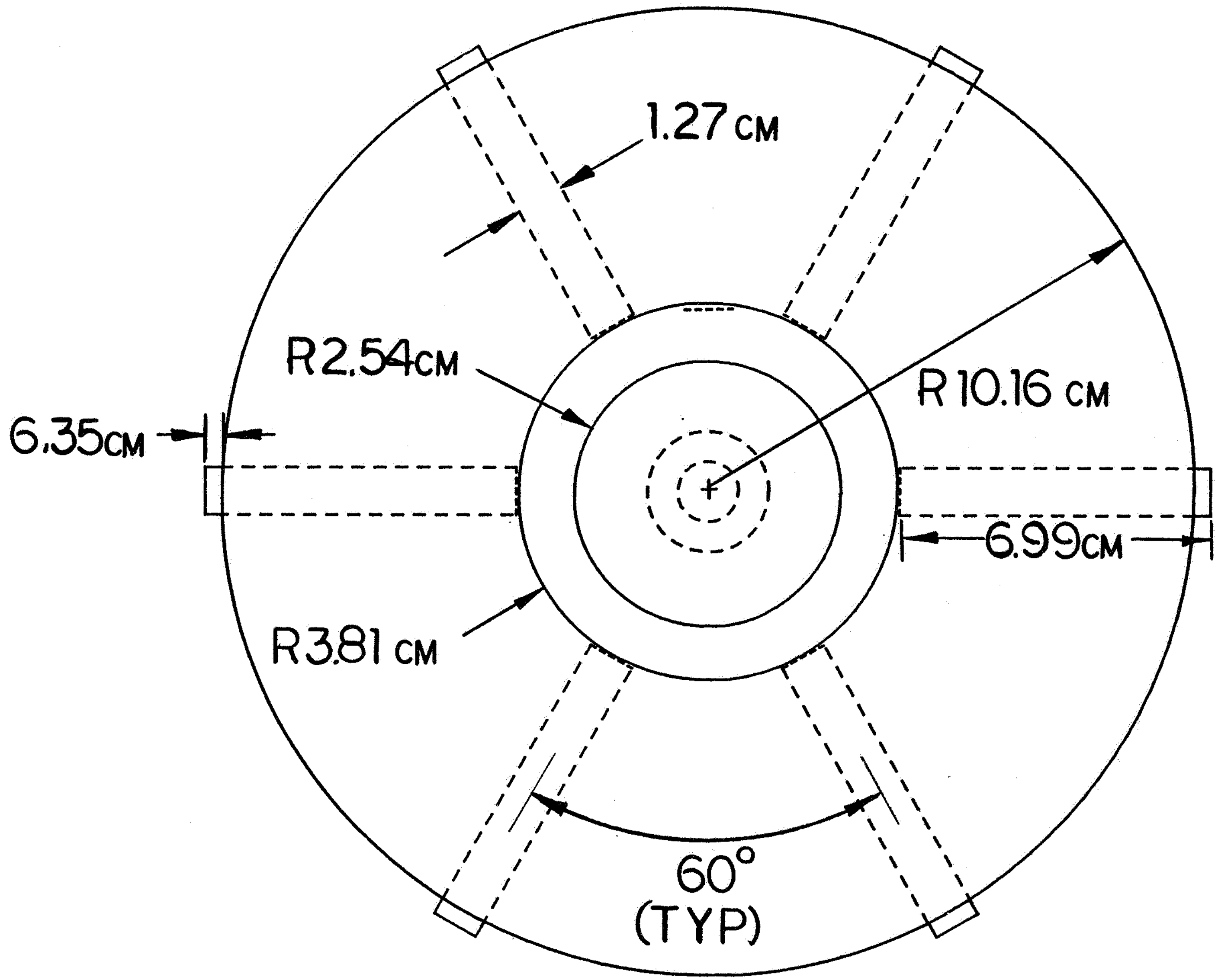
IMPELLER AND CASING FRONT VIEW



IMPELLER
(MOTOR SIDE)



IMPELLER
(RADIAL VIEW)



IMPELLER

(SUPPLY SIDE)

Appendix 4

Impeller

Force motor exerts on casing:

$$F = ma = (8.32\,kg)\left(\frac{9.8}{6}\ m/s^2\right) = 13.589\ N$$

4 rivets holding motor.

$$\text{Force on 1 rivet} = \frac{13.589N}{4} = 3.397N$$

Stress on rivet:

$$\sigma = \frac{F}{A}\ ;\ A = (\text{thickness of casing})\cdot(\text{diam. of rivet})$$

Beta Titanium rivets: $S_y = 176000\ \frac{lb}{in^2}\cdot\frac{0.4536\ kg}{lb}\cdot\frac{(1in)^2}{(0.0254m)^2}\cdot\frac{9.8}{6}\ m/s^2$

$$S_y = 202.11\ MPa$$

$$202.11\ MPa = \sigma = \frac{F}{A} = \frac{3.397N}{(0.00517m)(t)}$$

$$t \geq 3.251 \times 10^{-6}\ m$$

∴ Thickness of casing must be greater than 3.251×10^{-6}m to withstand the weight of the motor.

Make casing thickness 1 cm to be safe and allow for wear due to friction from the regolith.

Volume enclosed by casing:

$$V = \pi(11.43\text{ cm})^2(10.16\text{ cm}) + \tfrac{1}{2}(14.00\text{ cm} + 25.50\text{ cm})(10.00\text{ cm})(10.16\text{ cm})$$

$$V = 6176.60\text{ cm}^3$$

Outside volume of casing:

$$V = \pi(12.43\text{ cm})^2(12.16\text{ cm}) + \tfrac{1}{2}(15.00\text{ cm} + 26.50\text{ cm})(12.00\text{ cm})(12.16\text{ cm})$$

$$V = 8930.20\text{ cm}^3$$

Volume of casing attachment to bin:

$$V = \pi(4.00\text{ cm})^2(2.00\text{ cm}) + (5.00\text{ cm})(2.00\text{ cm})(7.62\text{ cm}) + (2.00\text{ cm})(2.50\text{ cm})(7.62\text{ cm})$$

$$V = 214.83\text{ cm}^3$$

Volume of material required for casing:

$$V = 8930.20\text{ cm}^3 - 6176.60\text{ cm}^3 + 214.83\text{ cm}^3$$

$$V = 2968.43\text{ cm}^3$$

Beta Titanium is chosen as the material.

Total mass of casing:

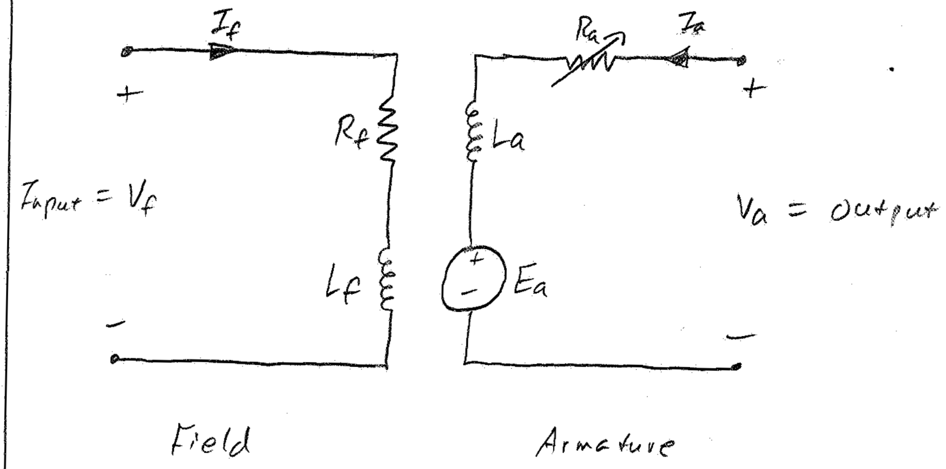
$$m = \rho V = (4.505\text{ g/cm}^3)(2968.43\text{ cm}^3) = 13372.78\text{ g}$$

Total mass of casing, impeller, and motor

$$m = 13.3723\text{ kg} + 9.0538\text{ kg} + 8.32\text{ kg}$$

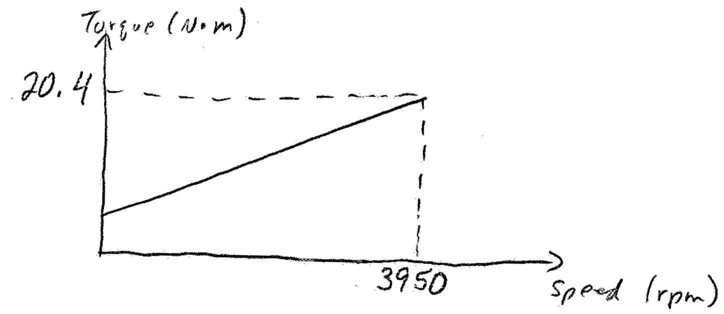
$$m = 30.746\text{ kg}$$

Schematic of SM 2447-01 Variable Speed D.C. Motor



The field accepts voltage from the source, and the armature delivers the necessary output voltage. The variable resistor R_a allows the motor to change speeds.

Characteristics:



Thermal Control:

Maximum temperature rise in casing due to friction = 100°C

Emittance of Aluminum: $\epsilon = 0.80$

Maximum temperature in motor:

$$180\,W \cdot 6\frac{°C}{W} = 1080°C$$

Even though the temperatures are well within the limits for titanium, the heat should be radiated away.

A material with high emissivity and low absorptivity is required to coat the casing and motor to radiate heat.

Use Fuller White Silicone paint:

absorptivity: $\alpha = 0.25$

emittance: $\epsilon = 0.90$

$$\frac{\alpha}{\epsilon} = \frac{0.25}{0.90} = 0.278$$

Bolts needed to attach impeller casing to bin:

Force exerted by casing $= ma = (30.746\,kg)\left(\frac{9.8}{6}\ m/s^2\right)$

$$F = 50.218\ N$$

Thickness of bin and plate attachment to casing:

$$t = (0.5\,in)(2.54\,cm) + 2.00\,cm = 3.27\,cm$$

Grip of bolt $= 3.27\,cm$ + nut thickness

$$= 3.27\ cm + 1.00\ cm = 4.27\ cm$$

5 mm Metric Grade Titanium bolts

diameter = 0.5 cm
Pitch = 0.8
Tensile Stress Area = $0.142\ cm^2$
Minor diameter = $0.127\ cm^2$
Proof Strength = 63.160 MPa   Endurance Limit = 21.130 MPa
Tensile Strength = 84.979 MPa
Yield Strength = 65.457 MPa

stiffness constant of bolt : $k_b = \frac{AE}{l} = \frac{\pi (0.005\,m)^2 (3.7896 \times 10^{10}\,Pa)}{(4)(0.0427\,m)}$

$$= 1.743 \times 10^7\ N/m$$

stiffness constant of member : $k_m = \dfrac{\pi E d}{2 \ln\left[\dfrac{5(l + 0.5d)}{l + 2.5d}\right]}$
(Aluminum bin)

$$= \frac{\pi (7.1484 \times 10^{10}\,Pa)(0.005\,m)}{2 \ln\left[\dfrac{5(0.0427 + (0.5)(0.005\,m))}{0.0427 + (2.5)(0.005\,m)}\right]}$$

$$= 6.399 \times 10^7\ N/m$$

Constant $C = \dfrac{k_b}{k_b + k_m} = \dfrac{1.743 \times 10^7}{1.743 \times 10^7 + 6.399 \times 10^7}$

$$C = 0.2141$$

$$\text{Force on bolts} = A_t S_{ut} - \frac{CnF}{2N}\left(\frac{S_{ut}}{S_e} + 1\right)$$

F = Force on bolts = 50.218 N

n = Safety factor = 3.0

N = number of bolts = ?

$$F_i = (1.42 \times 10^{-5} m^2)(84.979 \text{ MPa}) - \frac{(0.2141)(3.0)(50.218)}{2N}\left(\frac{84.979}{21.130} + 1\right)$$

$$F_i = 1206.702 - \frac{80.989}{N}$$

$$n = \left|\frac{A_t S_y - F_i}{CF}\right| = \left|\frac{(1.42 \times 10^{-5})(65.457 \text{ MPa}) - F_i}{(0.2141)(50.218)}\right|$$

$$n = \left|\frac{929.489 - F_i}{10.752}\right|$$

for $N = 3$ bolts: $F_i = 1.183$ kN

$$n = 8.78$$

Thus, the use of 3 bolts to attach the impeller casing to the bin is sufficient.

Welding of plate to bottom of bin:

Use edge weld: Best for sheet metal → bin material

Stress in weld $= \frac{F}{hl}$; F = force on weld (due to weight of casing)

h = weld throat

l = length of weld

$l = 0.02m$

$F = 50.218\ N$

h must be found

Need high strength for weld.

Use AWS E120 electrode : yield strength = 122.875 MPa

Stress in weld should not exceed yield strength:

$$h = \frac{F}{l\sigma} = \frac{50.218}{(0.02)(122.875\ MPa)}$$

$$h \geq 0.02043\ mm$$

Throat of weld should be 0.3 mm for added stress caused by vibration.

Force that weld can sustain:

$$F = hl\sigma = (0.0003m)(0.02m)(122.875\ MPa)$$

$$F = 737.25\ kN$$

Appendix 5

Bin

PRELIMINARY BIN DESIGN:

~~MATERIAL:~~	~~Density (g/cm^3)~~
~~TITANIUM~~	~~4.507~~
~~ALUMINUM~~	~~2.699~~

Titanium is 1.7 times heavier than Aluminum and since Aluminum meets the temperature and structural requirements, choose to design the bin out of Aluminum.

SEVERAL Strenghening mechanisms are available FOR Aluminum and ~~m~~ Aluminum Alloy ~~a~~ TABLE is given below:

MATERIAL	Tensile Strength (psi)	Yield Strength (psi)	Yield Strength (Alloy) / Yield Strength (Pure)
PURE AL	6,500	2,500	—
Solid Solution Strengthened	16,000	6,000	2.4
75% Cold Worked	24,000	22,000	8.8
AGE Hardened (5.6% Zn-2.5% Mg)	83,000	73,000	29.2

CHOOSING the AGE Hardened 7075-T6 alloy the density would equal .097 lb/in^3.

Preliminary Bin Design:

Material:

By using the Aluminum Alloy, the weight of the Bin would be equal to 293.246 lbs. If Titanium were used, the bin would weight approximately 489.685 Lbs. Thus, while not losing many of the benefits of Titanium's strength, the Aluminum would save an average of $4.3 million at 22,000 dollars per pound shipping weight. The bins weight will also be advantageous in balancing the fuel cell weight in the rear of the ARMS vehicle.

1st Description

→ The Bin was designed to allow a funneling of the regolith towards the pump inlet in the Rear. A Lip in the front of the Bin was designed to catch any thrown when soil which would otherwise go under the vehicle. The shape of the bin should allow for regolith thrown as high as 55° from 4 feet away or 71° from a 2 foot distance. Ideally, the soil should be tossed at an angle greater than 54° at a 2 foot distance.

Several Bin designs were considered and all but one were eliminated for:

Preliminary Bin Design

Not meeting the following criteria:

(1) Size - must be large enough catch regolith from a number of different angles; must not be too large.

(2) funneling - must be capable of funneling regolith towards pump - opening.

(3) Support - must be able to mount to the existing frame design easily.

(4) Weight - Having met the above criteria, the bin must be relatively light.

Thicknesses of the Bin should range from 1/16 of an inch in non-critical areas to 1/2 inch in stressed areas. The critical areas include the mounting plates on both sides of the bin, the pump mounting plate at the rear of the bin, and the bottom pan of the bin.

All Bin contours can be welded together with the entire unit bolted to the frame via the mounting plates. The same bolts used to mount the pump may be employed for mounting the bin as it will rest on the frame.

Preliminary Bin Design:

Material Weight:

front Lip: $4' * 1' * \frac{1}{16} * \frac{1}{12}$

BASE : $2' * 4' * \frac{1}{2} * \frac{1}{12} + 4' * 1' * \frac{1}{2} * \frac{1}{12}$

sides : $2\left(2' * 1\frac{1}{2}' * \frac{1}{16} * \frac{1}{12} + \frac{1}{2}' * 1' * \frac{1}{16} * \frac{1}{12} * \frac{1}{2} + 1' * 2.2' * \frac{1}{16} * \frac{1}{2} + .8' * 1' * \frac{1}{16} * \frac{1}{12} * \frac{1}{2}\right)$

TOP : $\frac{.5}{\sin 26} * 6' * \frac{1}{16} * \frac{1}{12} + \frac{.6}{\cos 51°} * 6' * \frac{1}{16} * \frac{1}{12} + 4' * \frac{1}{\cos 51°} * \frac{1}{16} * \frac{1}{12} +$

$\frac{1'}{\cos 51°} * 1' * \frac{1}{16} * \frac{1}{12} + 4 * 1 * \frac{1}{16} * \frac{1}{12}$

Support: $2(1' * 4' * \frac{1}{2} * \frac{1}{12})$

Back : $4' * 1.3' * \frac{1}{2} * \frac{1}{12} - \pi(10 * \frac{1}{12}')^2 * \frac{1}{2}'' * \frac{1}{12}$

Front Lip	:	.02083 ft^3
BASE	:	.500 ft^3
Sides	:	.17552 ft^3
TOP	:	.468304 ft^3
Support	:	.33333 ft^3
Back	:	.125764

$$\boxed{\text{Total material volume} = 1.749512\ ft^3}$$

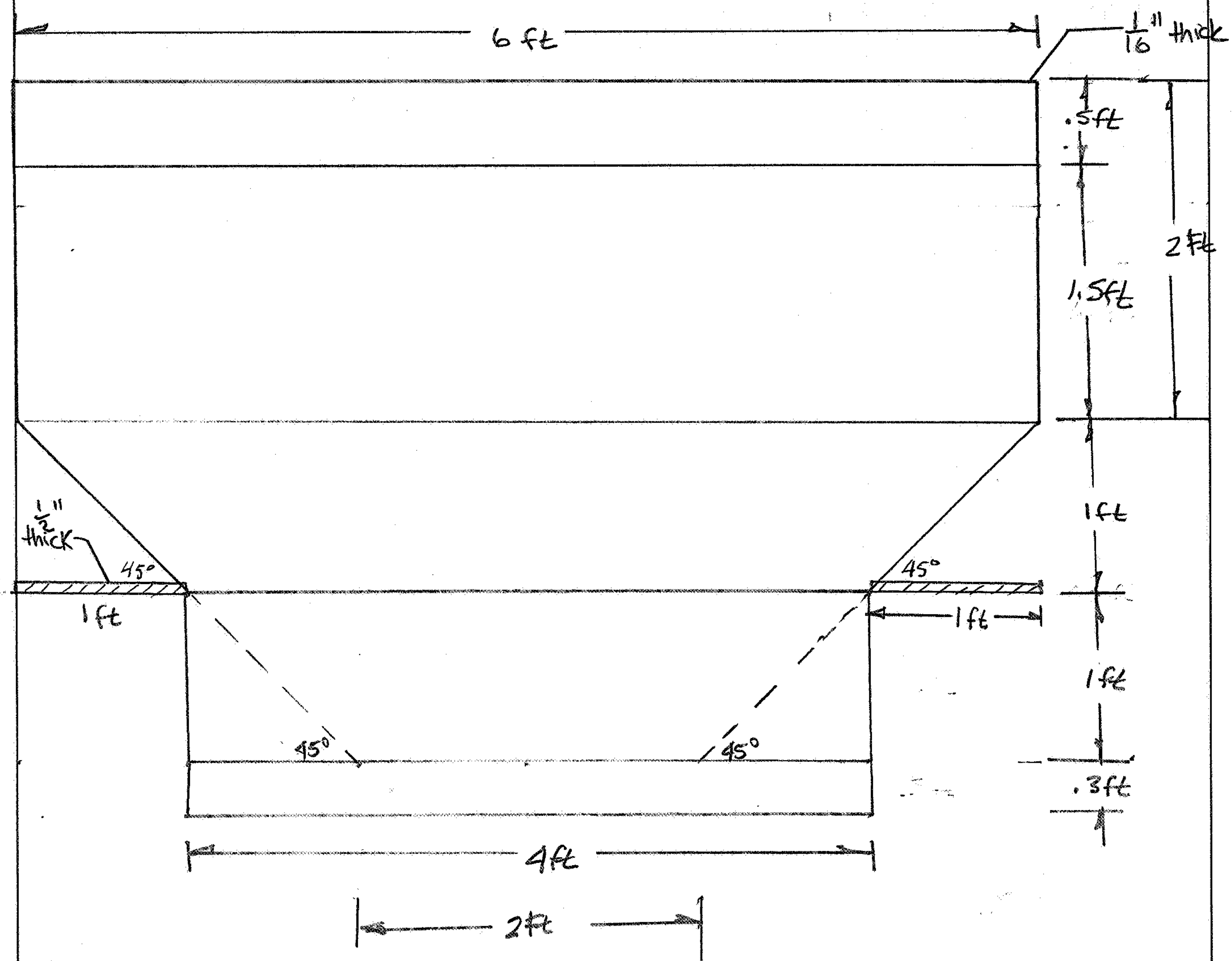
Weight = material volume * material density

$$\text{Weight} = 1.749512\ ft^3 * .097 \frac{Lb}{in^3} * \left(\frac{12\ in}{ft}\right)^3$$

$$\boxed{\text{Weight} = 293.246\ Lb}$$

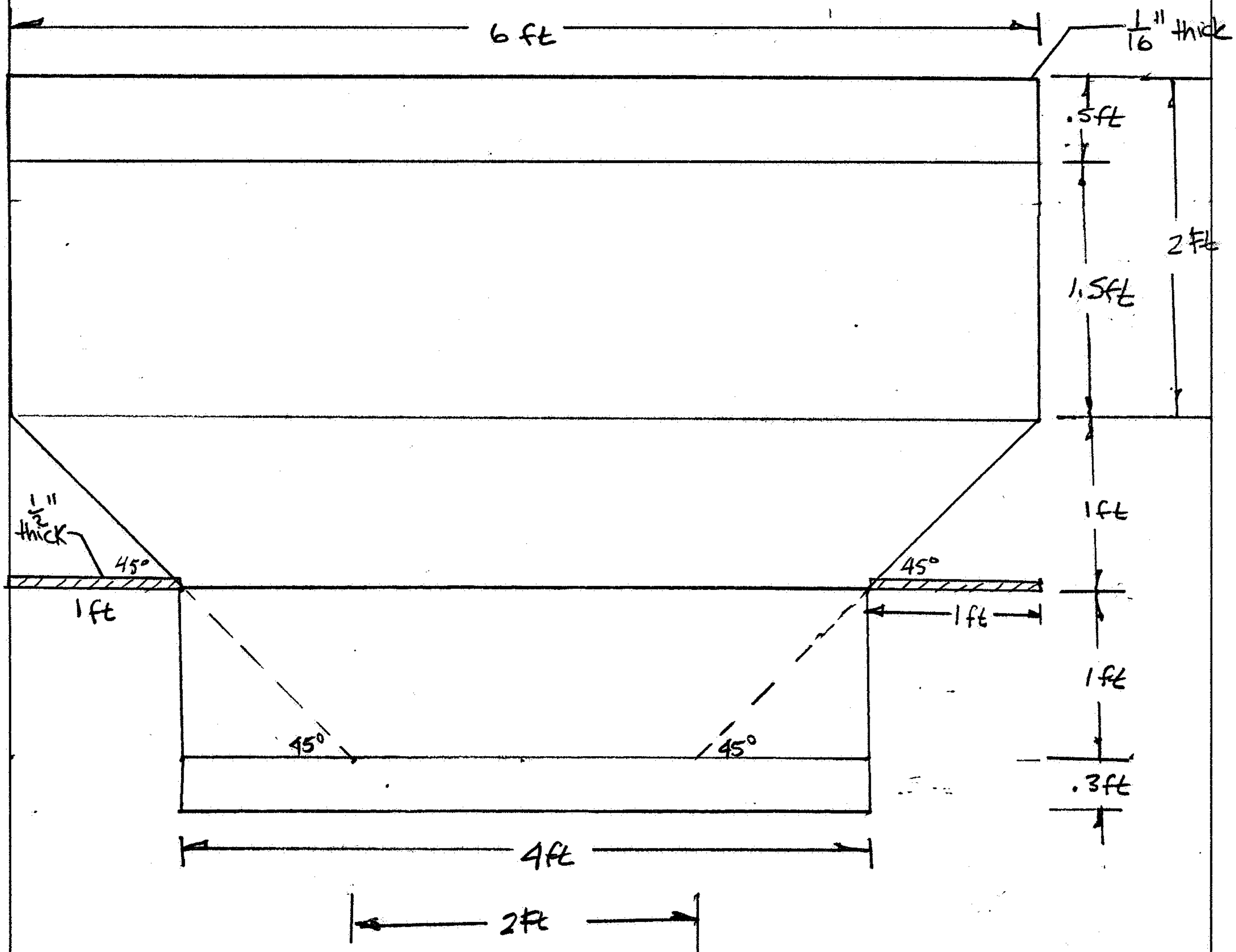
Preliminary Bin Designs

FRONT VIEW:

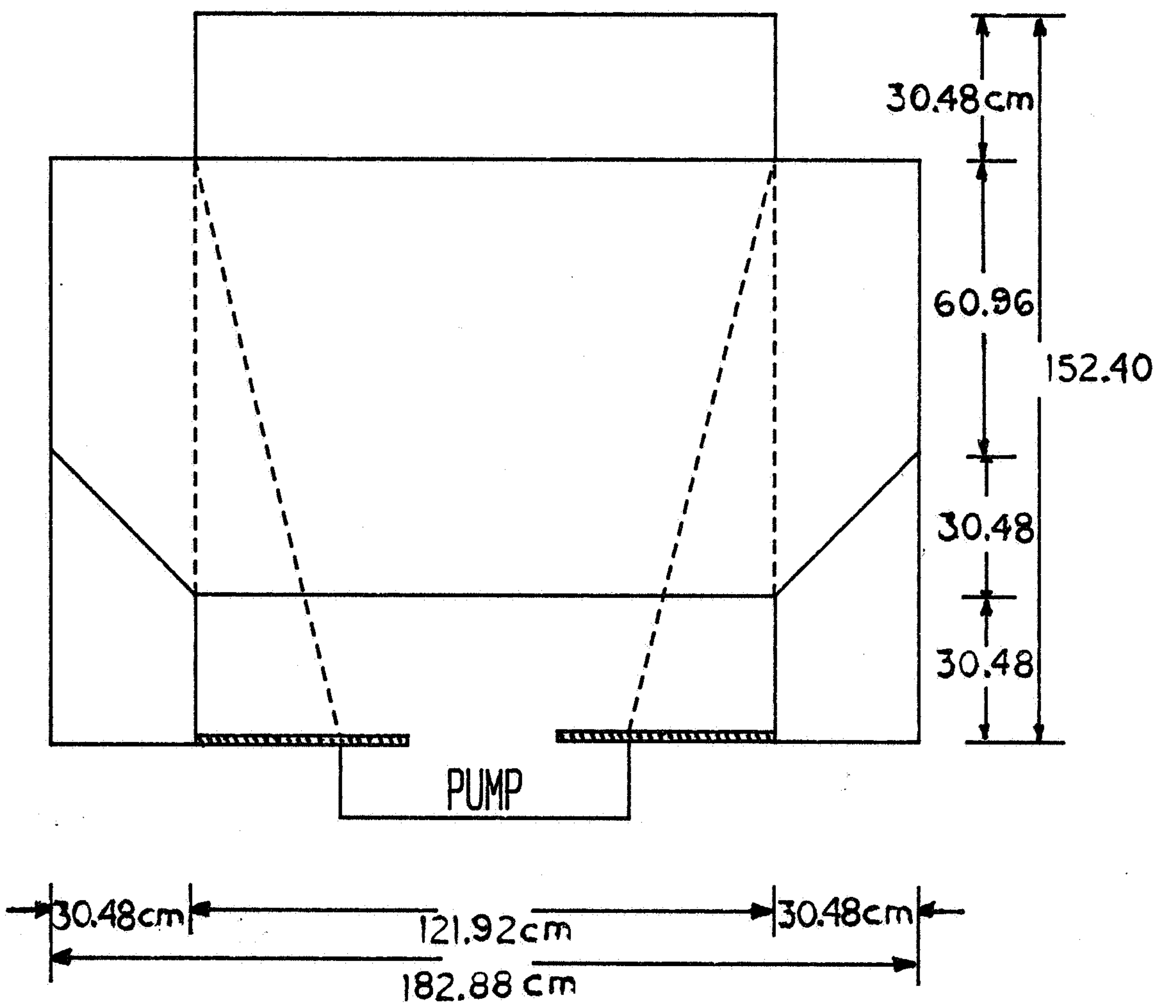


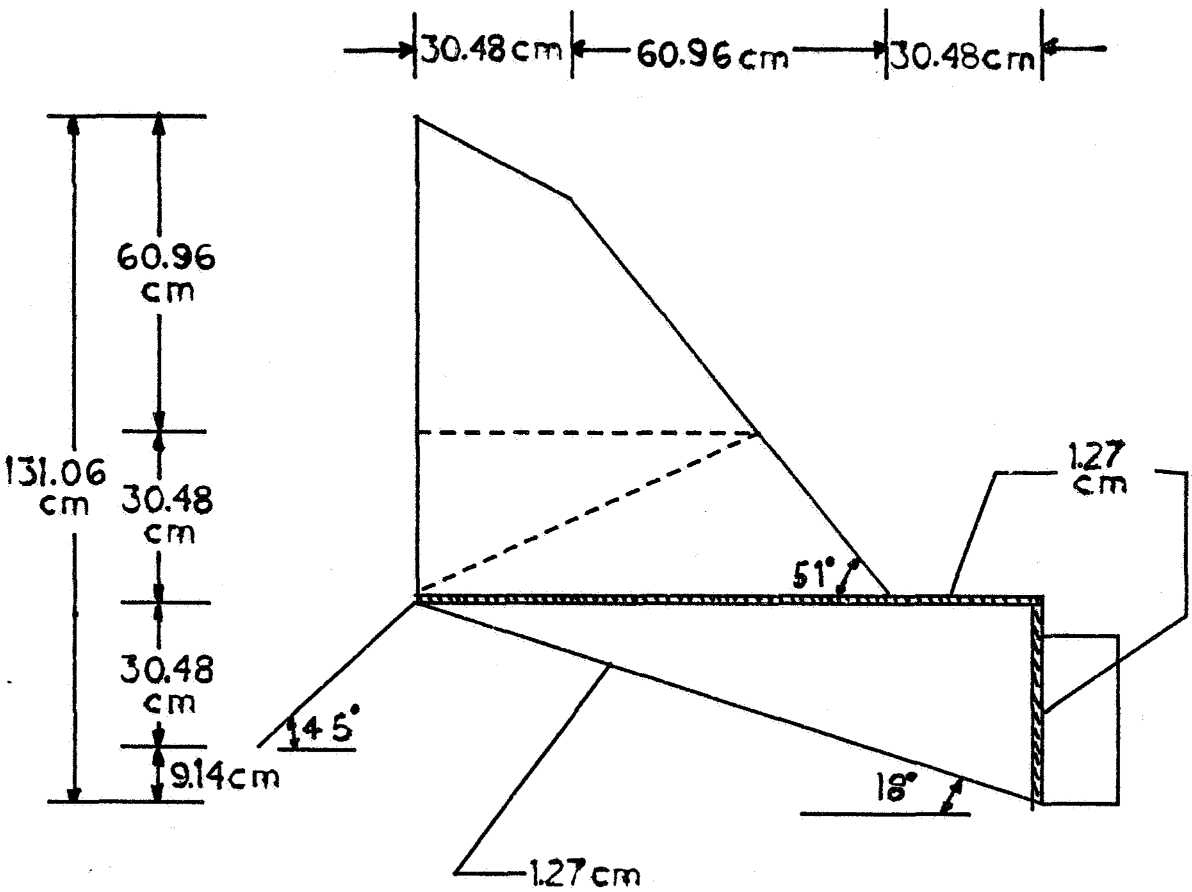
Preliminary Bin Designs

FRONT VIEW:



BIN TOP VIEW


30.48cm
60.96
152.40
30.48
30.48
PUMP
30.48cm
121.92cm
30.48cm
182.88 cm



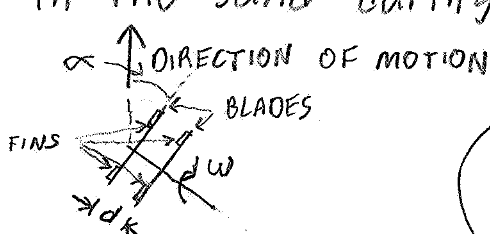
BIN SIDE VIEW

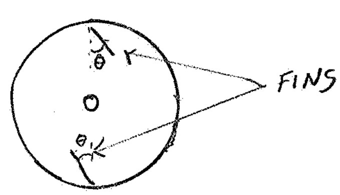
Appendix 6

Operation

To test the soil collection disks for the Automated Regolith Movement System a prototype was built. This prototype was made from old 7 3/4 in. circular saw blades fitted with fins to throw the soil. The blades were then bolted to a 1/2 in. threaded rod shaft which had one end turned down to fit into a standard 3/8 in. drill chuck. This setup created a versatile test device that could be powered by an ordinary hand drill.

The testing consisted of a series of qualitative tests while varying the operating conditions. The tests took place in a sandy area to simulate the lunar regolith. The blade spacing (d), angle of attack (α), fin angle (θ), and angular velocity (ω) of the disks were varied. The approximate angle that the sand was thrown at as well as the sorting ability of the disks was recorded. Various sized rocks were placed in the sand during testing.





The desired result of operation was to have the sand thrown up with a trajectory of 60° while the rocks were not thrown. By varying the speed of rotation, this could be obtained. At very low speeds, the sand did not gain enough momentum to be thrown. At slightly higher speeds, the sand would be thrown almost vertically. At speeds of roughly 200-600 rpm, the desired effect was achieved. When the blades encountered rocks, the rocks would be thrown straight back with a trajectory of less than 5°.

To test different fin angles, the fins on one blade were attached at an angle of 20° This fin angle did not significantly alter the trajectory of the sand. However, this blade tended to throw small rocks up, thus defeating the purpose of the blades.

The blades worked well at an angle of attack varying from 0° to 45°. At angles greater than 45°, the sorting ability of the blades greatly diminished allowing for small rocks to be thrown.

The spacing of the blades did not alter the performance of the blades. With the blades spaced at either 1", 2", or 3" apart, each blade acted independently of the other. Rocks did not get lodged in between the blades but the assembly tended to walk over rocks large enough to get stuck.

This experiment verified the throwing and sorting ability of a finned disk. A more in depth experiment using more disks and multiple shafts could be undertaken to determine torque requirements and test mechanism designs.

Disk Design

MASS TO BE MOVED:

$$\frac{29{,}000\ ft^3}{} \left| \frac{.0283\ m^3}{1\ ft^3} \right. = 821.2\ m^3 = 821{,}200\ L$$

TIME CONSTRAINT:

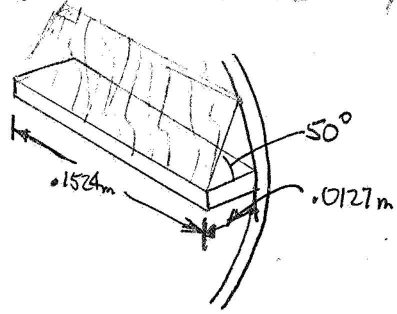
12 days = 1036 800 sec

MASS FLOW RATE = $\frac{821{,}200\ L}{1036\,800\ sec} = .792\ L/sec$

FIN DIMENSIONS

1) Avg DEPTH OF HARDPAN ≃ 6 IN = .1524 m
2) NO PARTICLE LARGER THAN ½"
IS TO BE THROWN IN HOPPER .0127 m

MAX. VOL DISPLACED BY FIN:



$$\begin{aligned} Vol &= \tfrac{1}{2} b h l \\ &= \tfrac{1}{2}(.0127)(.0127 \tan 50°)(.1524) \\ &= 1.465 \times 10^{-5}\ m^3 \\ &= .01465\ L \end{aligned}$$

MAX VOL BY DEVICE:

$$\frac{10\ \text{DISK}}{1\ \text{REV}} \left| \frac{2\ \text{FIN}}{1\ \text{Disk}} \right| \frac{.01465\ L}{1\ \text{FIN}} = .293\ L/\text{REV}$$

RPM OF DISK:

$$\frac{(RPM)}{60} \times \frac{.293\ L}{REV} = L/s$$

$\dot{m}$ (L/s)	RPM
.488	100
.733	150
.977	200
1.22	250

CHOOSE RPM OF 200, FIN WILL NOT DISPLACE ITS MAX. VOL. AT ALL TIMES.

MASS OF DISK ASSEMBLY

SHAFT DIA. & DISK THICKNESS TAKEN FROM DISK HARROW WITH SAME DIA. DISK.

d_s (SHAFT DIA) = 1.375" T_d (DISK THICKNESS) = .1875" d_d = 18.00"

L_s (SHAFT LENGTH) = $(32'' - 5T_d) = \boxed{31.062''}$

FIN VOL = $(6'' \times .5'' \times .125'') = .375 \text{ IN}^3$

Disk Vol = $\pi R^2 T_d = = \pi (9.0)^2 (.1875) = \underline{47.713 \text{ IN}^3}$

SHAFT Vol = $\pi R^2 L_s = \dfrac{\pi (1.375)^2}{4} (31.062) = 46.124 \text{ IN}^3$

TOTAL VOL

T.Vol = 10 FINS + 5 DISK + SHAFT.

$= (10)(.375) + 5(47.713) + (46.124)$

T.Vol = 288.439 IN^3

$= \underline{1.760 \times 10^{-3} \text{ m}^3}$

WEIGHT ON EARTH = Vol × UNIT WEIGHT (CARBON STEEL) = $(1.760 \times 10^{-3} \text{m}^3)(76.5 \text{ kN/m}^3) = \underline{134.63 \text{ N}}$

MASS = $\dfrac{\text{WEIGHT}}{g} = \dfrac{134.63 \text{ N}}{9.81 \text{ m/s}^2} = \underline{13.72 \text{ kg}}$ OF CARBON STEEL / GANG

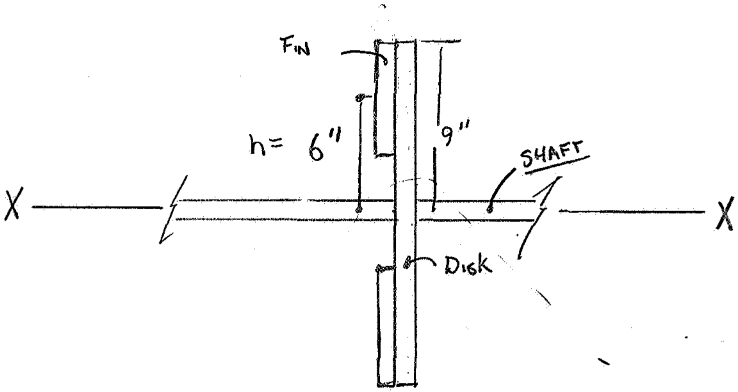
MASS OF SHAFT = Vol × ρ = $\dfrac{46.124 \text{ IN}^3}{} \Big| \dfrac{1 \text{ m}^3}{1.638 \times 10^5 \text{ IN}^3} \Big| \dfrac{76.5 \times 10^3 \text{ N}}{\text{m}^3} \Big| \dfrac{s^2}{9.81 \text{ m}} = \boxed{2.20 \text{ kg}}$

ρ of CARBON STEEL = 7798.2 kg/m^3

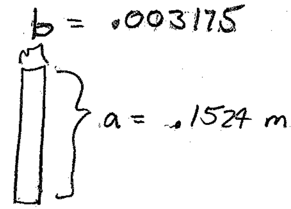
MASS OF DISK = $\dfrac{47.713 \text{ IN}^3}{} \Big| \dfrac{1 \text{ m}^3}{1.638 \times 10^5 \text{ IN}^3} \Big| \dfrac{7798.2 \text{ kg}}{\text{m}^3} = \boxed{2.27 \text{ kg}}$

MASS OF FIN = $\dfrac{.375 \text{ IN}^3}{} \Big| \dfrac{1 \text{ m}^3}{1.638 \times 10^5 \text{ IN}^3} \Big| \dfrac{7798.2 \text{ kg}}{\text{m}^3} = \boxed{.0179 \text{ kg}}$

INERIAL MOMENTS



MASS MOMENT



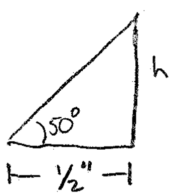
FIN: $I_x = \frac{m}{12}(a^2+b^2)$

$$I_x = \frac{.0179}{12}(.1524^2 + .003175^2)$$

$I_x = 3.467 \times 10^{-5}\ kg \cdot m^2$ ⇐ FIN ONLY

FIN COMPLETELY LOAD:

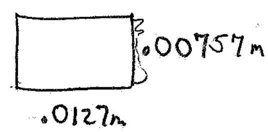
REGOLITH SHAPE



$h = \tan 50° (.0127m)$

$\underline{h = .0151\,m}$

=



VOL. OF REGOLITH

$$(.00757\,m)(.0127\,m)(.1524\,m) = \boxed{1.465 \times 10^{-5}\ m^3}$$

MASS OF REGOLITH

$$M = Vol \times \rho = (1.465 \times 10^{-5}\ m^3)\left(2400\ \frac{kg}{m^3}\right) = \underline{.0352\ kg}$$

$$I_x = \frac{m}{12}(a^2+b^2) = \frac{.0352(kg)}{12}(.1524^2 + .00757^2)$$

$I_x = 6.830 \times 10^{-5}\ kg \cdot m^2$ ⇐ REGOLITH

DISK: $I_x = \frac{md^2}{8} = \frac{(2.27\,kg)(.4572\,m)^2}{8} = .0593\ kg\,m^2 \Leftarrow$ Disk

SHAFT: $I_x = \frac{md^2}{8} = \frac{(2.20\,kg)(.0349\,m)^2}{8} = 3.35 \times 10^{-4}\ kg\,m^2 \Leftarrow$ SHAFT

TOTAL MASS MOMENT

$$I_{xx} = 5I_{x_{DISK}} + I_{x_{SHAFT}} + (I_{x_{FIN}} + I_{x_{REGO}} + m_{FIN}h^2 + m_{REG}h^2)10$$

$$I_{xx} = 5(.0593) + (3.35\times10^{-4}) + 10(3.467\times10^{-5} + 6.830\times10^{-5} + .1524^2(.0179 + .0352))$$

$$\boxed{I_{xx} = .310\ kg\,m^2}$$

POWER

$P = \frac{1}{2}I_{xx}\omega^2$ $\quad$ RPM = 200 $\quad$ $\omega = \frac{2\pi(200)}{60} = 20.94$

$P = \frac{1}{2}(.310\ kg\,m^2)(20.94\ \text{RAD/sec})^2 = 68\ W$

$P = 1.34\,\frac{Hp}{kW}(.068\ kW) = .091\ Hp \simeq .1\ Hp$

$T_{@200\,RPM} = \frac{63000\,P(Hp)}{n(RPM)} = \frac{63000(.1)}{200} = 31.5\ lb\text{-}in$

TOTAL POWER & TORQUE REQUIREMENTS

$P = .2\ Hp$ $\quad$ $T_{@200\,RPM} = 63.0\ in\cdot lb$

(5)

DUE TO THE EXTREMELY SLOW FORWARD VELOCITY OF THE MACHINE; TWO MORE FINS WERE ADDED TO EACH DISK & THE RPMS INCREASED TO 500 RPM.

RECALCULATIONS:

$$I_{xx} = 5I_{DISK} + I_{SHAFT} + 20(I_{FIN} + I_{REGO} + m_{fin}h^2 + m_{REG}h^2)$$

$$I_{xx} = 5(.0593) + 3.35\times10^{-4} + 20(3.467\times10^{-5} + 6.830\times10^{-5} + (.0352 + .0179)(.1524)^2)$$

$$I_{xx} = .324 \text{ kg m}^2$$

POWER FOR ONE GANG (5 DISKS):

$$P = \tfrac{1}{2}I\omega^2 \qquad \text{RPM} = 500 = 52.36 \text{ RAD/sec}$$

$$P = \tfrac{1}{2}(.324 \text{ kg m}^2)(52.36 \text{ rad/sec})^2$$

$$= 443.5 \text{ W}$$

$$P = (.4435 \text{ kW})(1.34 \text{ Hp/kW}) = .594 \text{ Hp} \simeq \underline{.6 \text{ Hp}}$$

$$T_{@500} = \frac{63000 \text{ Hp}}{\text{RPM}} = \frac{63000(.594)}{500} = \underline{74.9 \text{ lb-in}}$$

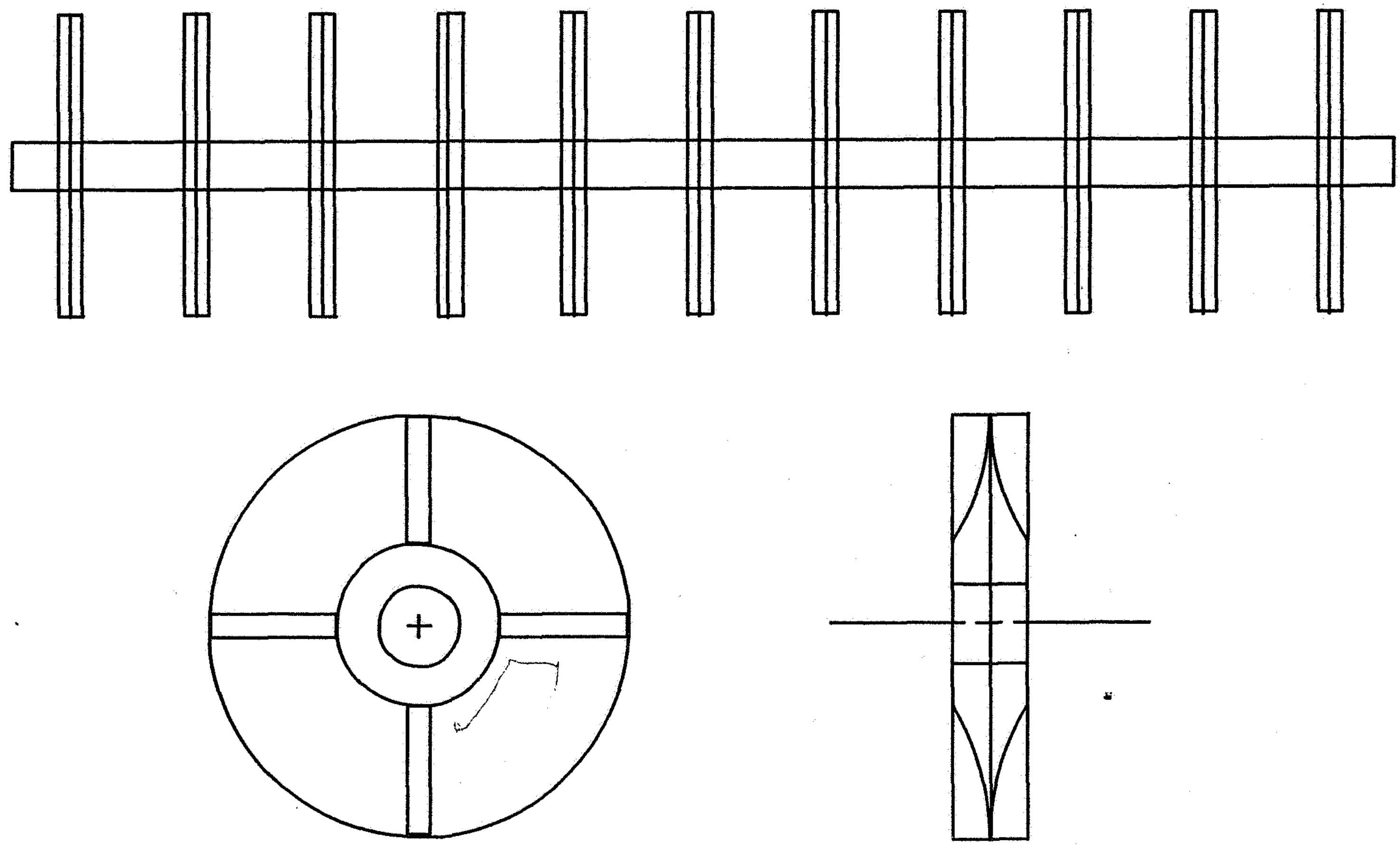
NEW MASS FLOW RATES:

$$\frac{10 \text{ Disk}}{1 \text{ Rev}} \left| \frac{4 \text{ Fin}}{1 \text{ Disk}} \right| \frac{.01465 \text{ L}}{1 \text{ Fin}} \left| \frac{8.33 \text{ Rev}}{1 \text{ sec}} \right. = 4.88 \text{ L/sec}$$

TIME CONSTRAINT:

$$\frac{(821,200 \text{ L})}{(4.88 \text{ L/sec})} = 168,277 \text{ sec} \simeq \underline{2 \text{ DAYS}}$$

REGOLITH SIFTING & THROWING SYSTEM



Appendix 7

Design Alternatives

Puncture Analysis on the Cylinder

The cylinder was assumed to be made out of standard Al at 1/16" thick. The method used is very conservative therefore when the force is equal to the yield Strength/$\sqrt{3}$ the cylinder will be slightly damaged. A small test was performed to determine if values followed actual results.

Calculations:

Damage to cylinder:

$$\frac{Y.S}{\sqrt{3}} = F \quad \text{where} \quad F = \frac{m\,\Delta V}{\Delta t} \qquad \Delta t = \frac{\text{thickness}}{V_i/2}$$

→ Y.S.: Yield Strength of Al = 9,676,246.63 N/m^2

→ m: mass of object hitting cylinder

- $m = 1/6\,\pi\,D^3\,\rho$

Note: Assumed object hitting the cylinder is a sphere.

m = (Vol Sphere) Density

(D)iameter of sphere = this valued is varied.

ρ = Density of Al at Moon gravity = 2,900 kg/m^3

→ ΔV: Change in Velocity: $V_i - \cancel{V_f}^{0} = V_i$

→ Δt: $\dfrac{t}{(V_i/2)}$

t: thickness of Al = 1/16"

Calculations:

Damage to cylinder → $\frac{Y.S.}{\sqrt{3}} = F$ where $F = \frac{m \Delta V}{\Delta t}$

$\Delta t = \frac{\text{thickness of Al}}{Vi/2}$

a) Y.S. ≡ Yield Strength of Al = 9,676,246.63 N/m²

b) m ≡ mass of object hitting cylinder

1.) m = (Volume Sphere)(Density)

$= (\frac{1}{6}\pi D^3)(\rho)$

Not: Assumed object hitting the cylinder is a sphere

2) ρ ≡ Density of regolith at moon conditions

$\rho = 24000$ kg/m³

c) ΔV ≡ Change in velocity

1) Vi ≡ initial velocity of porticals leaving pump.

2) Vo ≡ impact velocity assumed to be zero

d) Δt ≡ t ≡ thickness of Al. $t = \frac{1}{8}$"

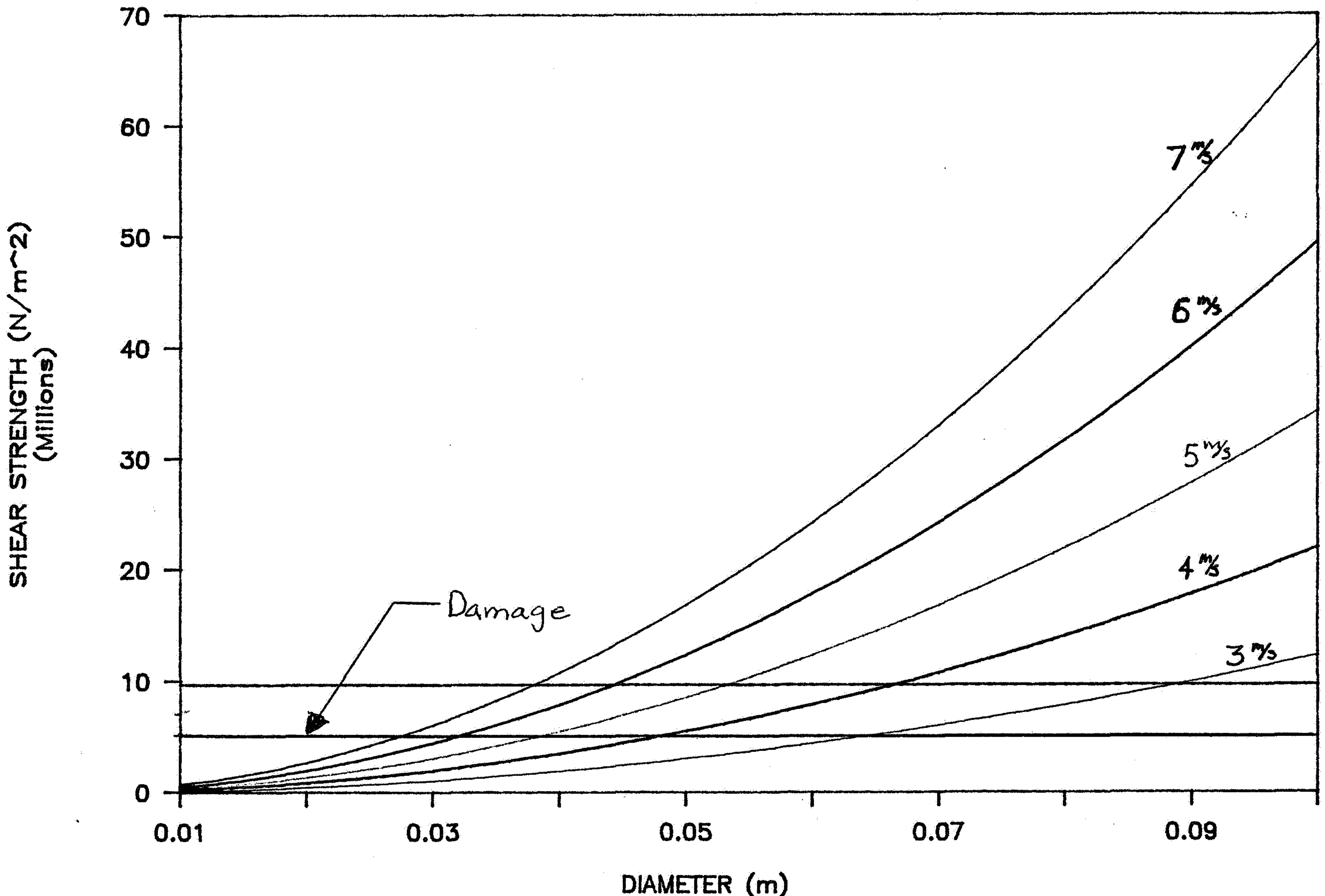
$Vi/2$ ≡ mean velocity

$\Delta t = \frac{t}{Vi/2}$

The calculation were don of Lotus 123 by varying the velocity and the diameter of the regolith. - (See Graph)

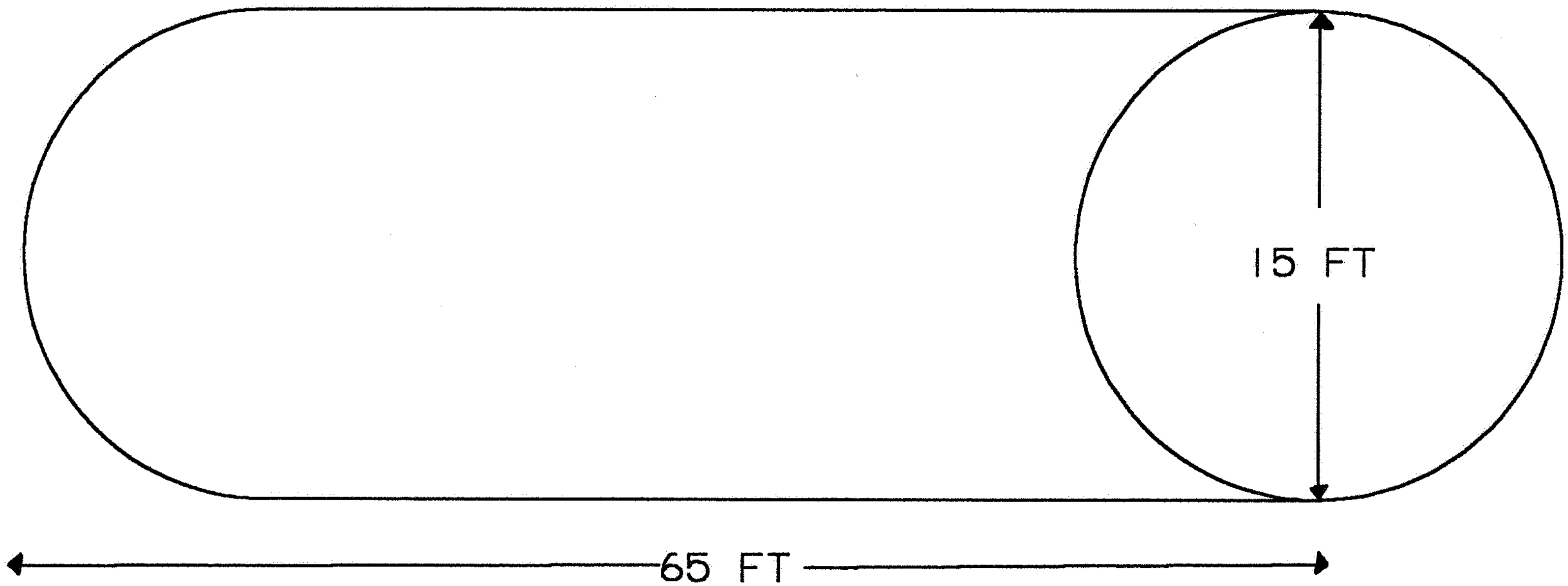
PUNCTURE ANLYSIS ON THE CYLINDER

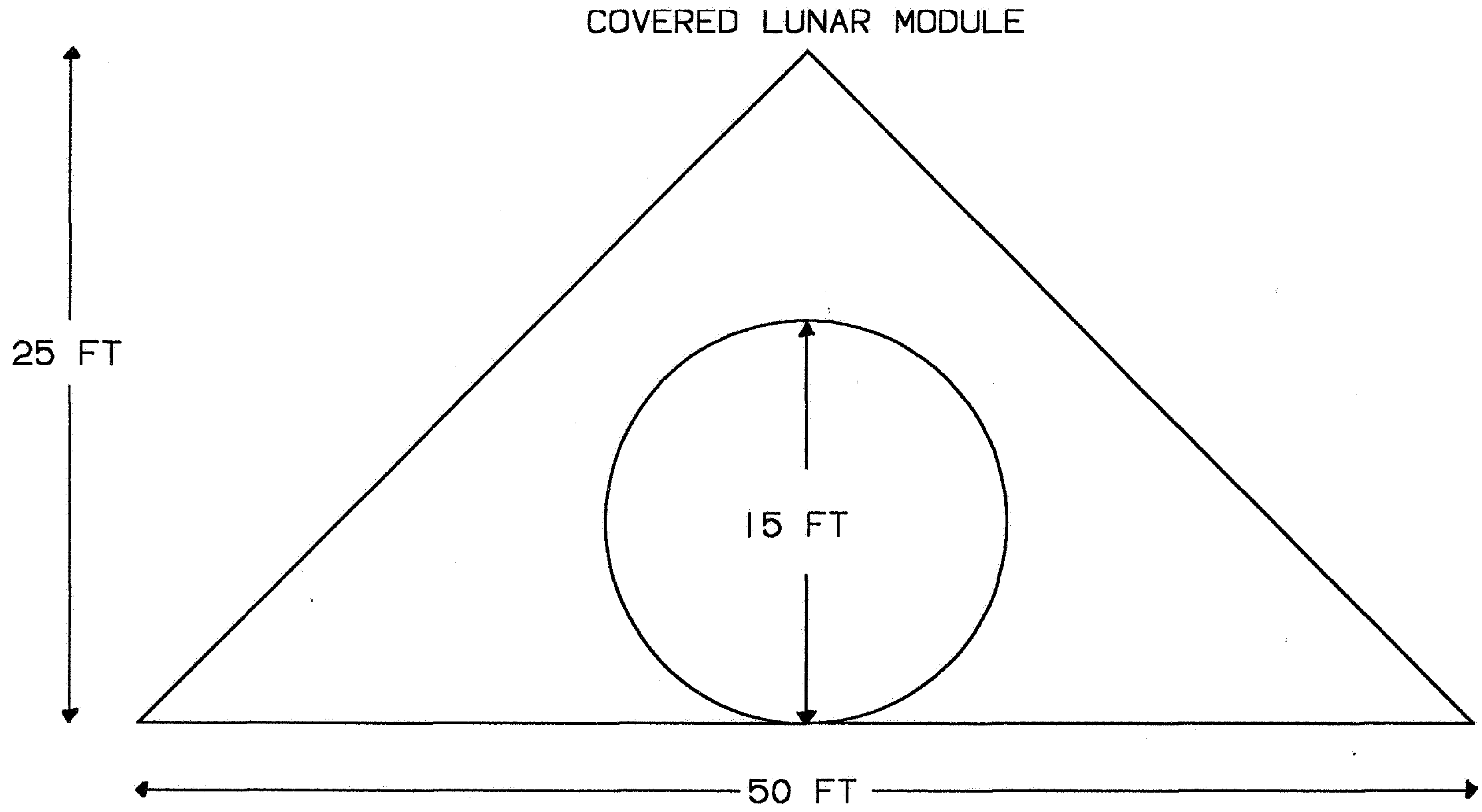
ME 4182 ARMS DESIGN PROJECT FALL,1987



— VELOCITY = 5 (m/s²) —— VELOCITY = 6 (m/s²)

LUNAR MODULE

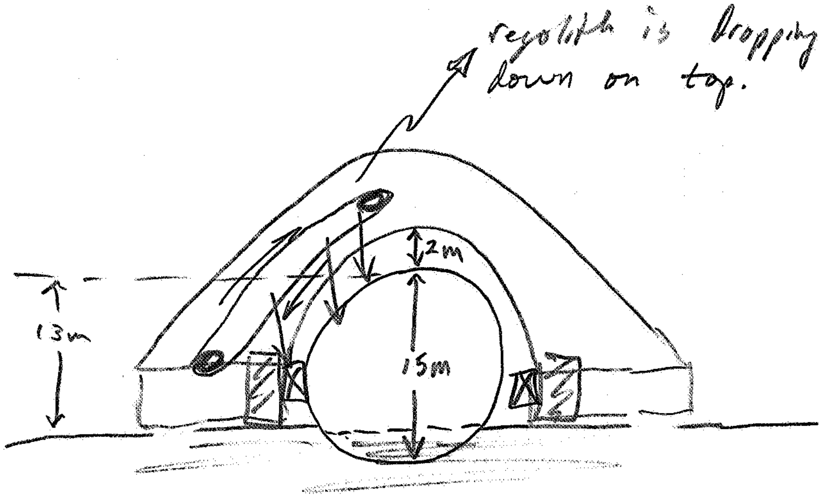


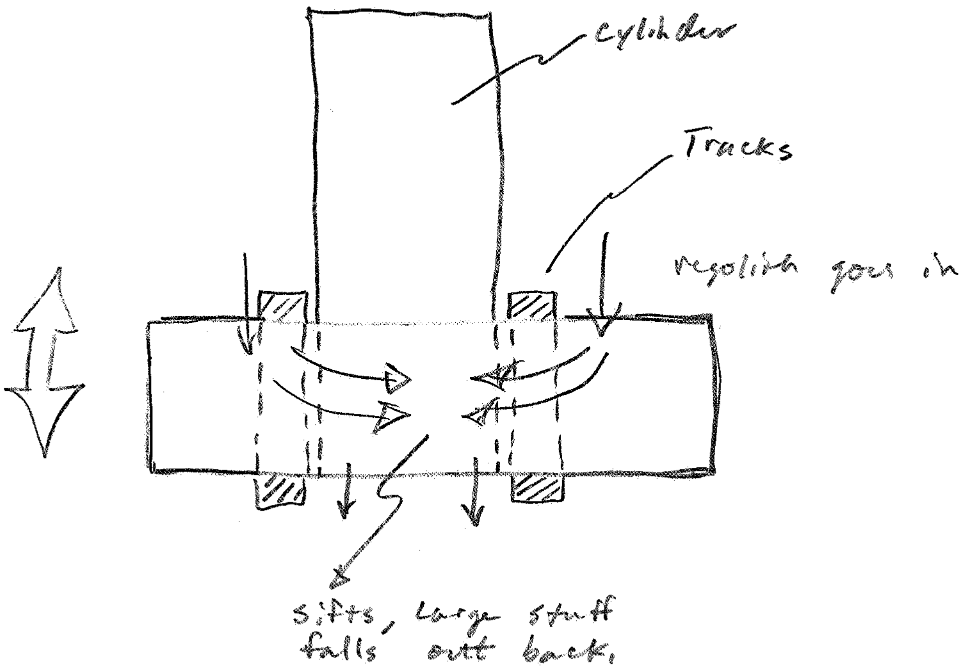


VOLUME OF REGOLITH

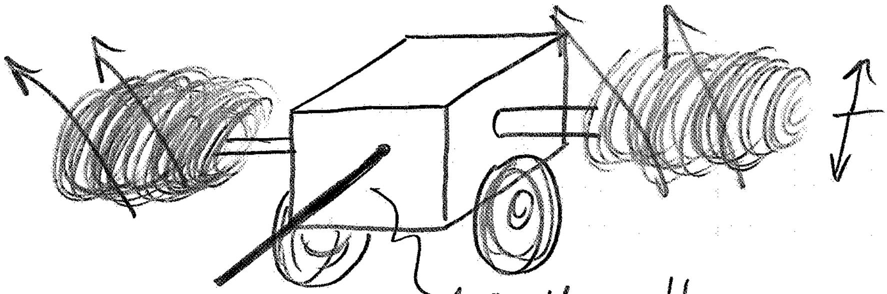
$$\left(25^2 - \pi(7.5)^2\right)65 = 29{,}138\ ft^3$$

$$\frac{29138\ ft^3}{23\ ft^3/yd.} = 1080\ yds^3 \Rightarrow 77$$

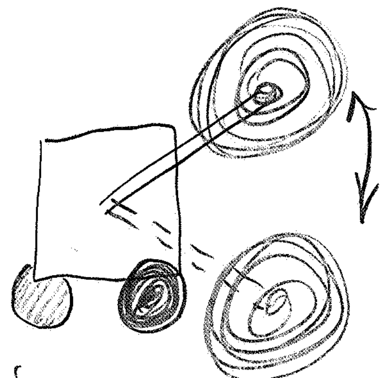




– Vehicle moves forward and backward until cylinder is covered.

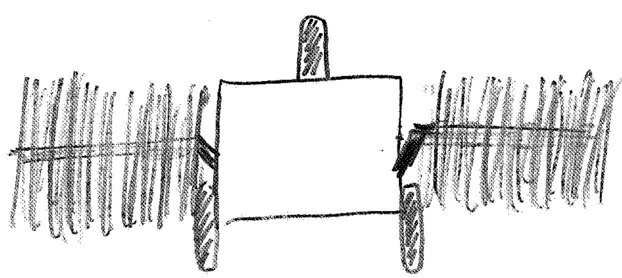


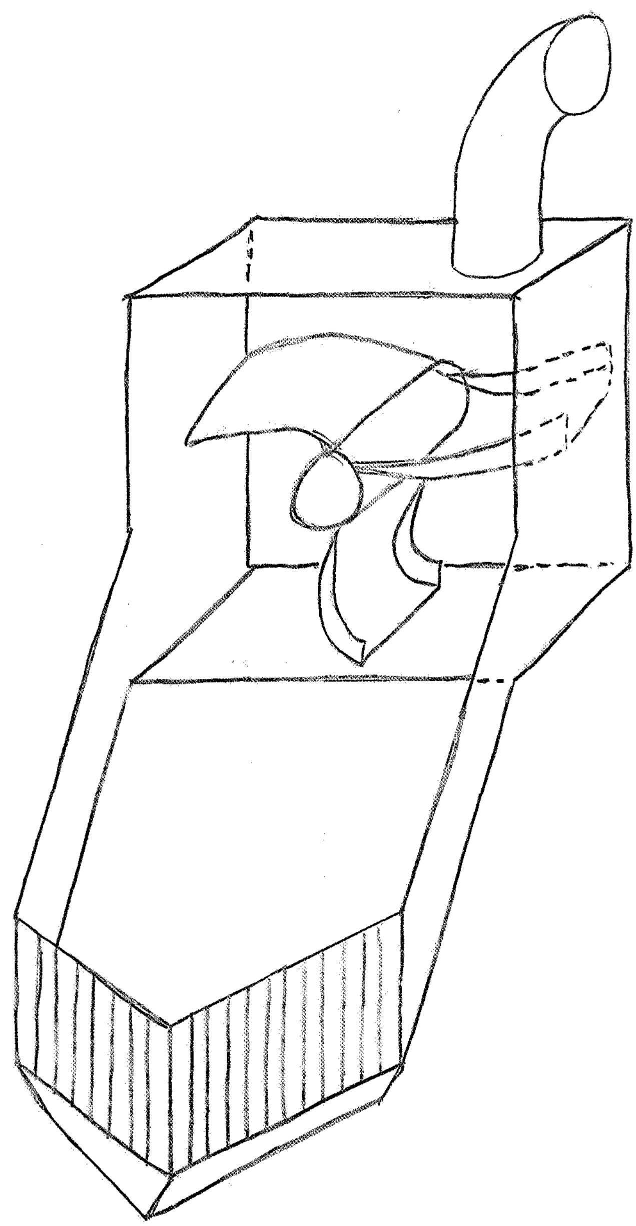
Brushes on both sides, movable up and down.
small, compact three wheeled vehicles.

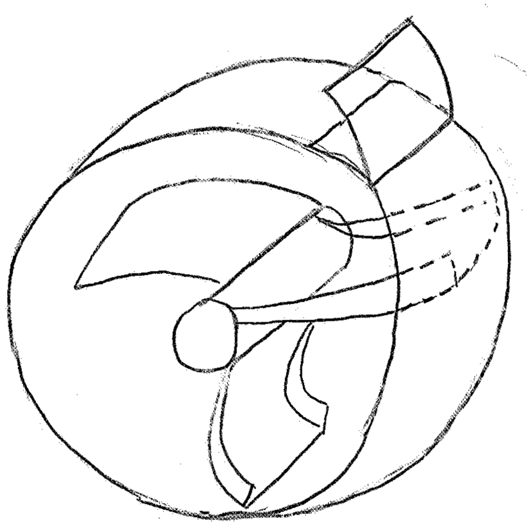


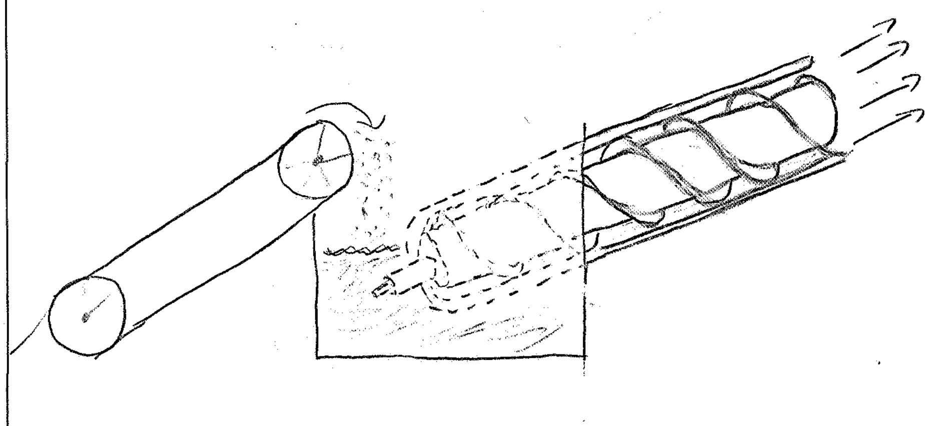
- Solid tires

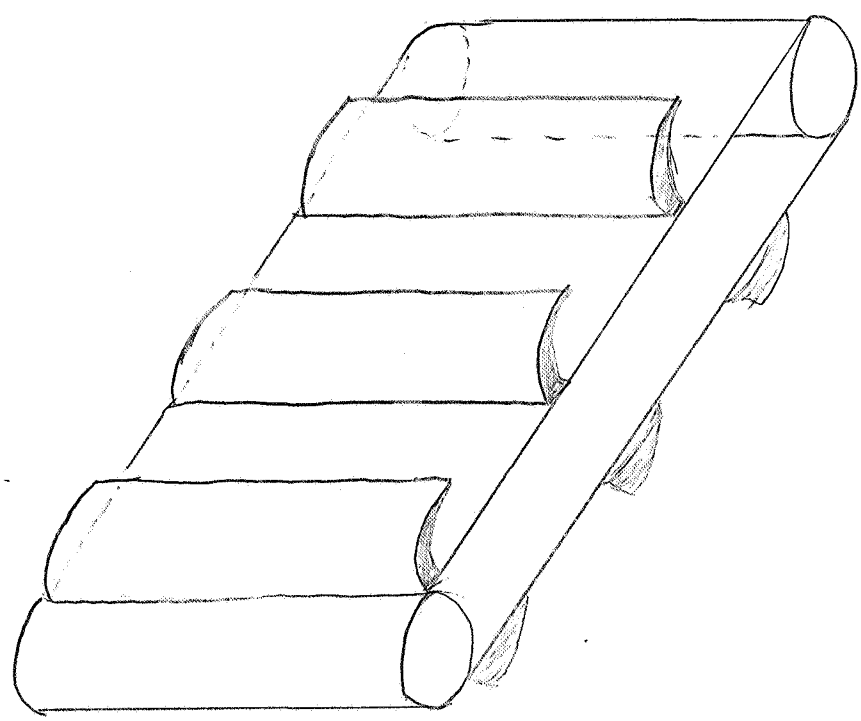
- rechargeable batteries.



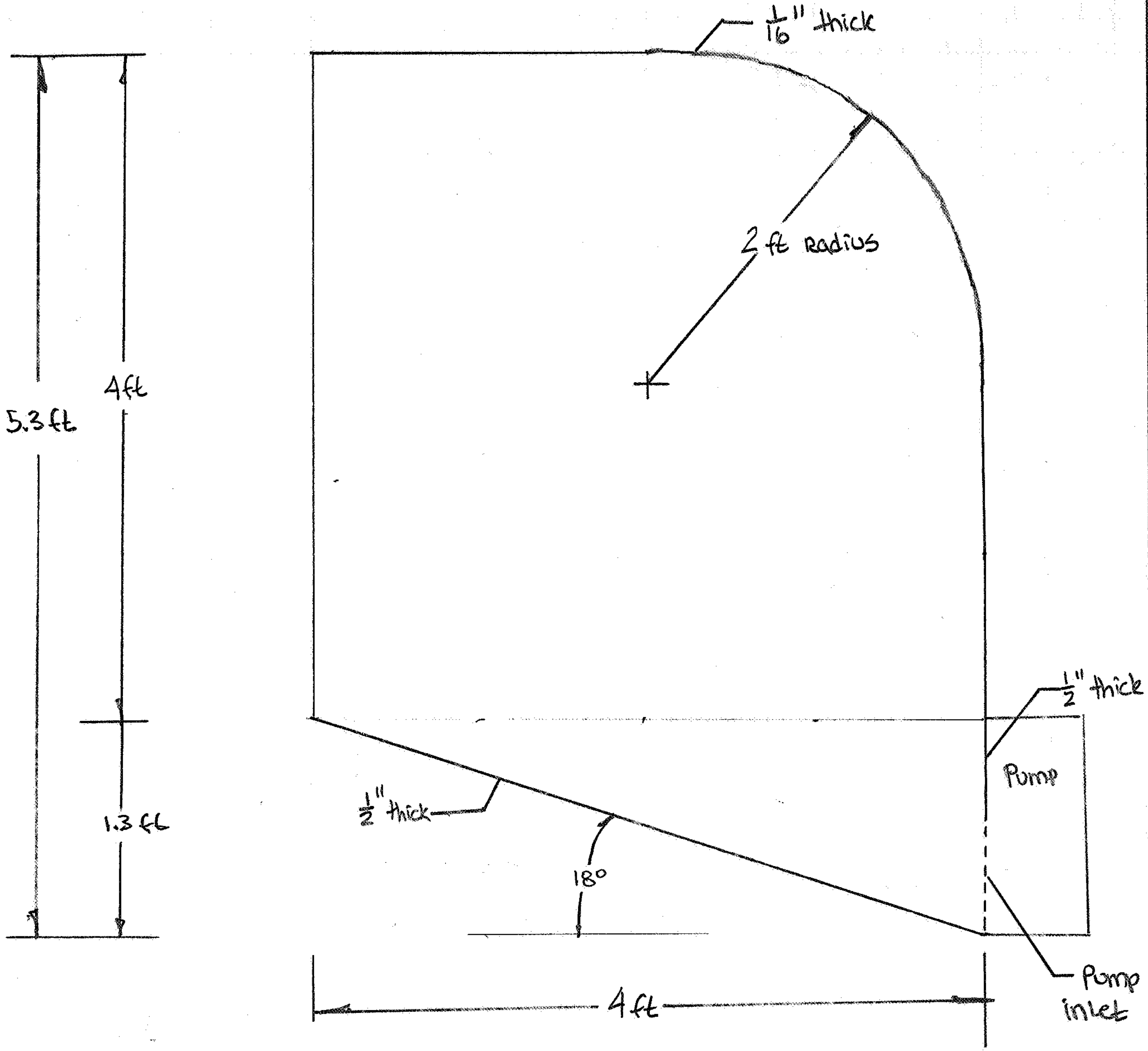








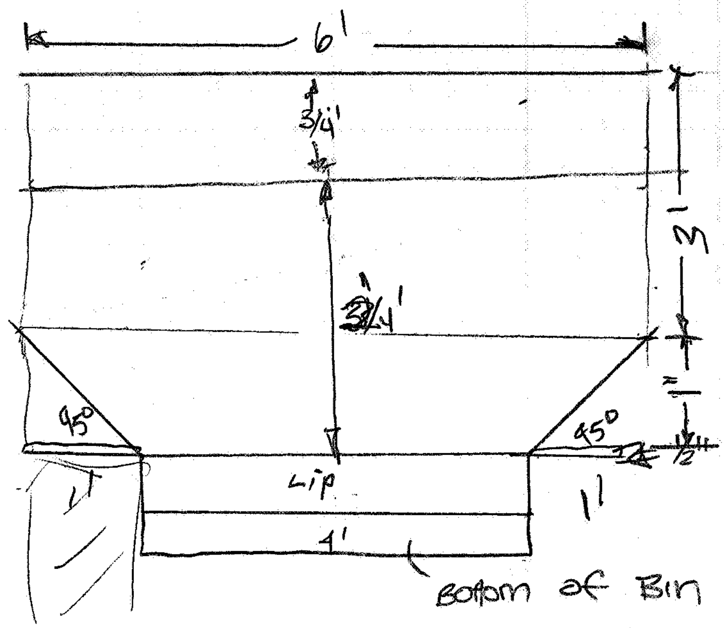
Preliminary Bin Design

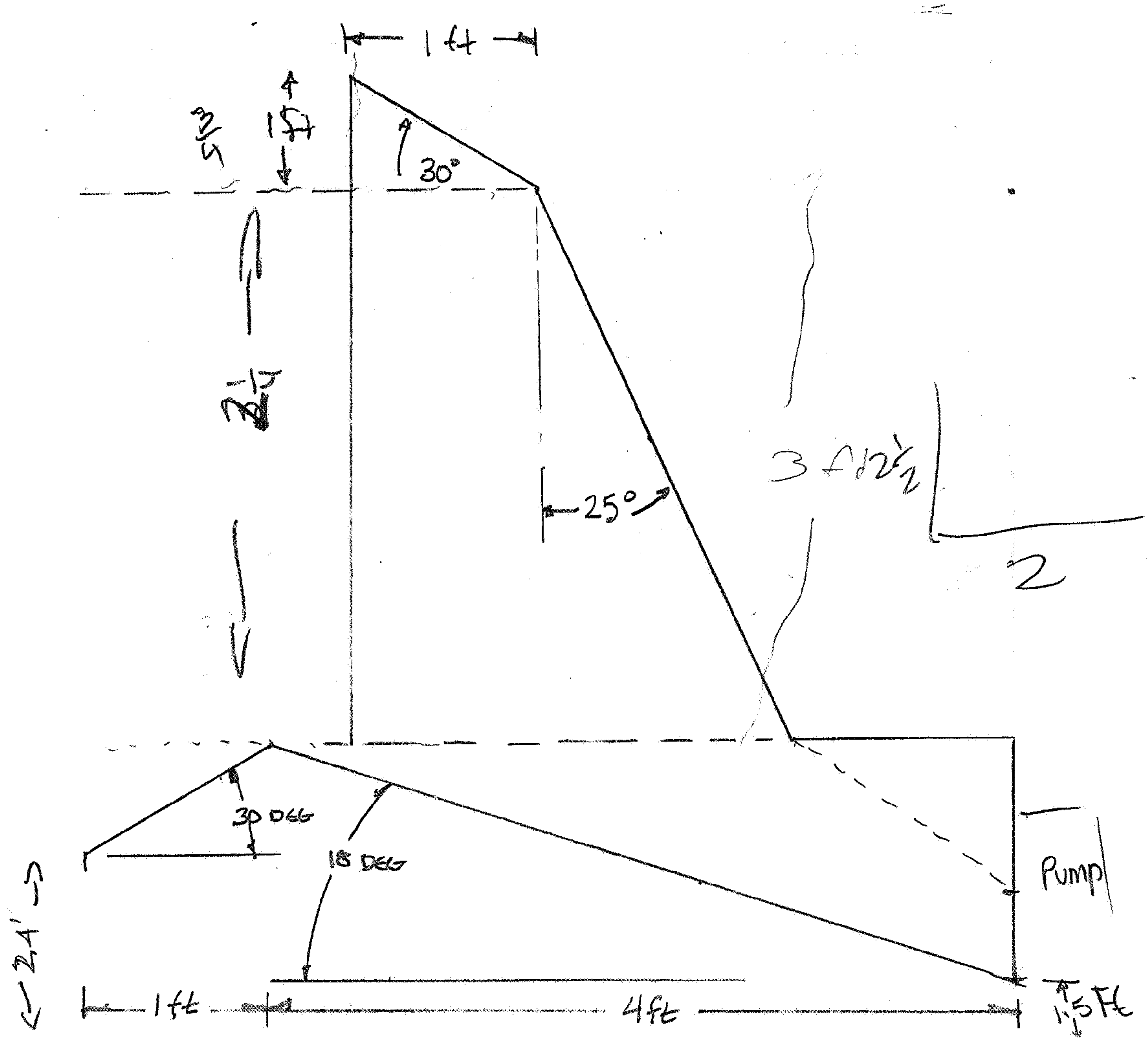


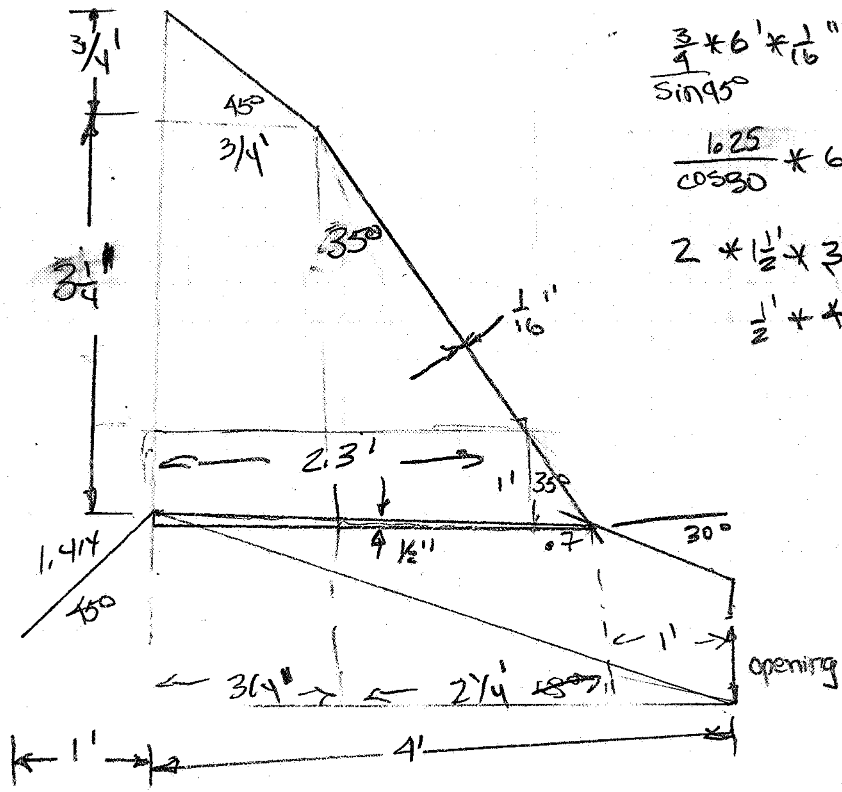
* found to have too much material and did not facilitate funneling thus eliminated design.

BIN DESIGN

MATERIAL: ALUMINUM = 2.7g/cm³







$$\frac{\frac{3}{4} * 6' * \frac{1}{16}''}{\sin 45°} =$$

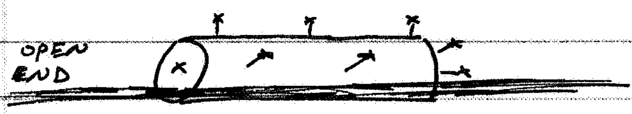
$$\frac{1.25}{\cos 30} * 6' * \frac{1}{16}'' =$$

$$2 * 1\tfrac{1}{2}' * 3' * \tfrac{1}{2}'' =$$

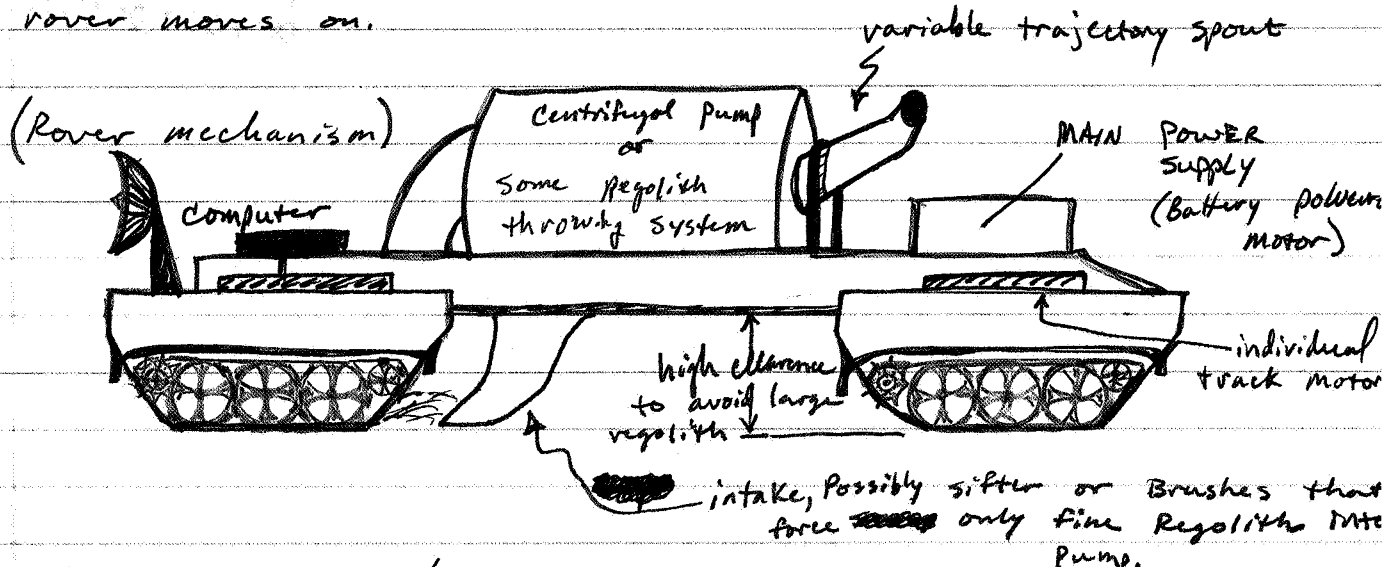
$$\tfrac{1}{2}' * 4'$$

A.R.M.S.

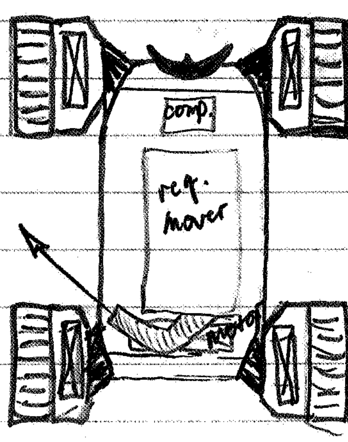
HOW DOES IT FIND AND ORIENT ITSELF TO THE CANNISTER?

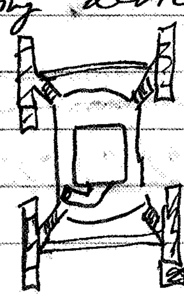


DEPENDING on the way it will be buried, transducers arranged all over the cannister will give off signals to the rover. The rover will take bearings on it to find it. Then as it buries it, Photocells on the transducers will turn off when buried. When all signals are off rover moves on.



- Four treads, Battery/Electric motor driven
- on-Board computer for signal differentiation
- Regolith moving system in the middle, centrifugal pump with a scoop in front feeding a sifting device first, then the pump.



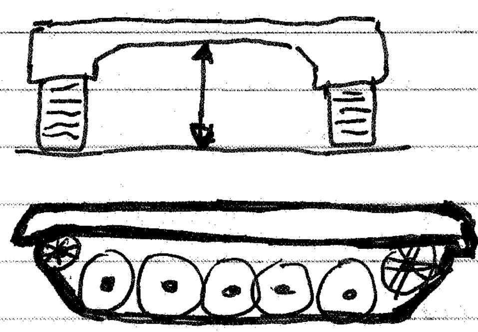


- very loose suspension with a maximum clearence depending on area terrain. Similar to Dune buggy

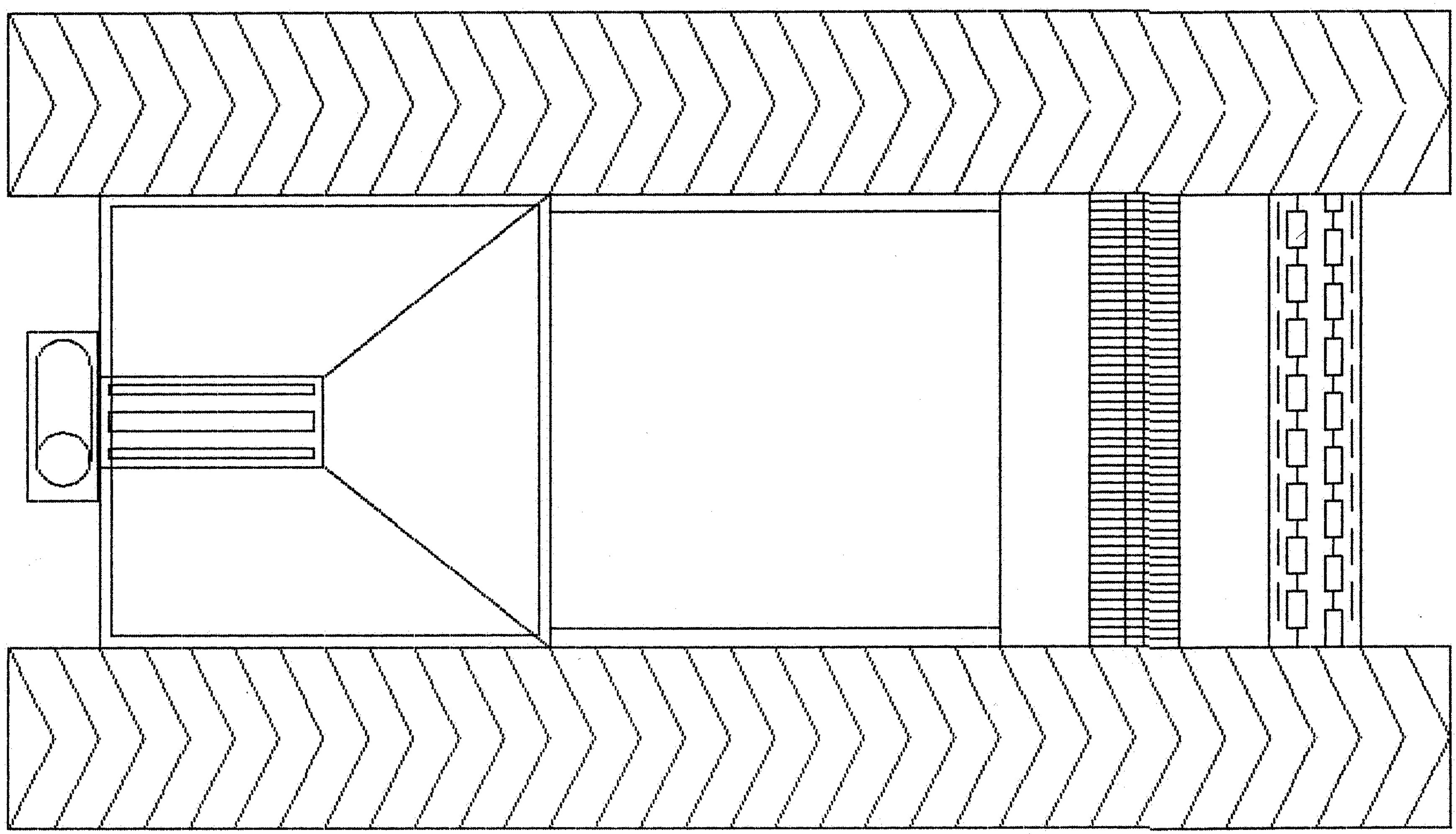
MM

120 A.

ORIGINAL PAGE IS
OF POOR QUALITY



A.R.M.S.
(Conceptual)



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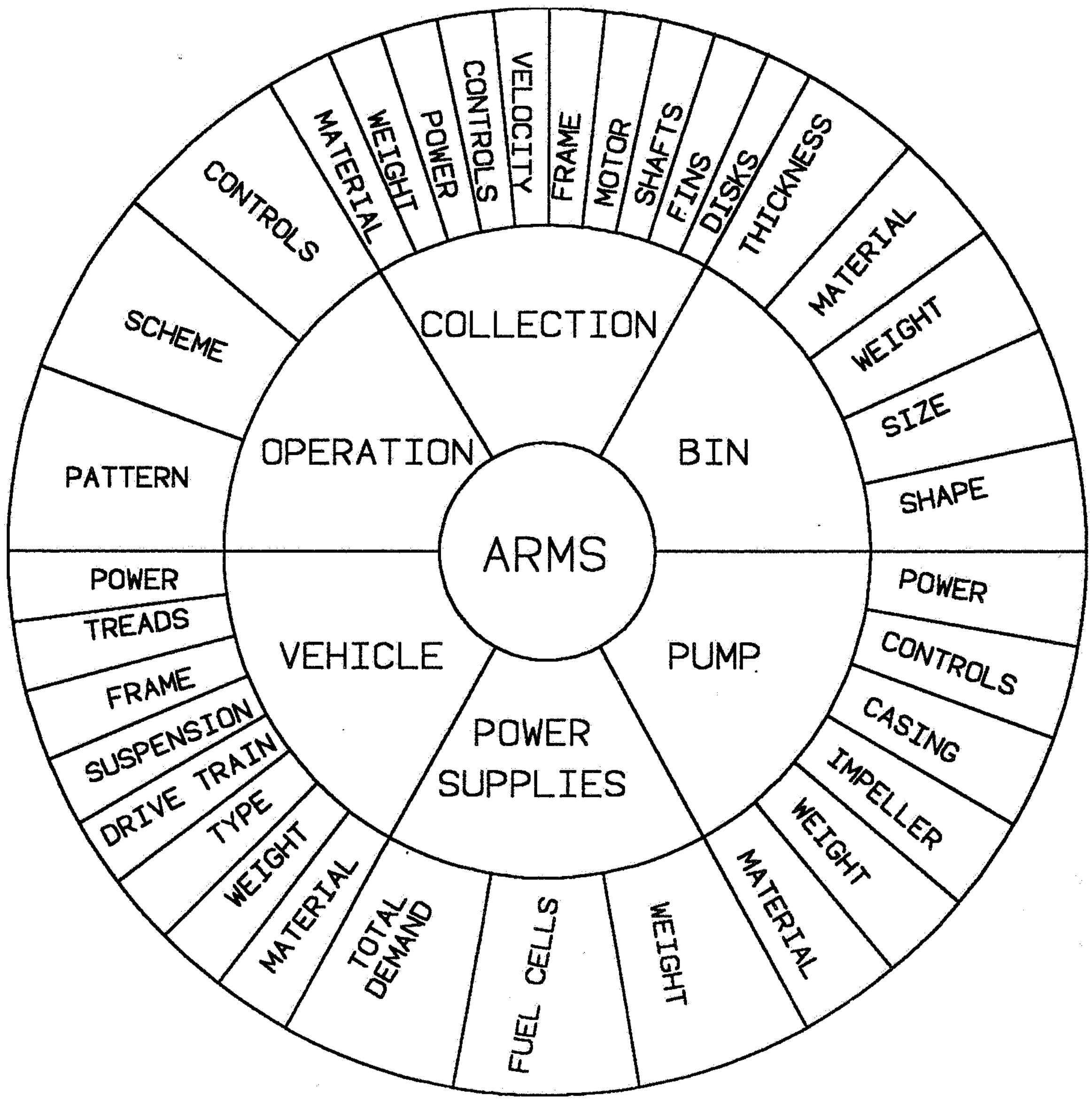
Appendix 8

Decision Matrix

DECISION MATRIX



Appendix 9

Progress Reports

PROGRESS REPORT GROUP3 OCTOBER 1, 1987 The project was given a title: Automated Regolithe Movement System. Initial requirements of the design were evaluated and a problem statement was formulated. Group members are currently researching other projects and formulating preliminary designs.

Progress Report

Week 2

p. The group formulated a problem statement and decided on a system to cover the modules. Each individual is currently researching the different requirements of the design.

1. Kates, Andy Wrote final problem statement. In the process of contacting sources at The University of Houston to find out more about the modules.

2. Shaw, Chris- Researched patents on augers and snowblowers for design ideas. Worked on blower design.

3. Stevens, Anthony- Researched patents on augers and snowblowers. Worked on blower design.

4. Hagler, Ben- Worked on blower design. In the process of learning Apollo solid modeling system.

5. McWhorter, Valerie- Worked on blower design. Made graphics of module. Investigating computer graphics systems.

6. Dimarco, Mike- Researched lunar surface conditions.

7. Porter, Tim- Did lunar trajectory calculations. Wrote program to calculate velocity to range relationships.

8. Crawford, Paul- Worked on lunar trajectory calculations.

Progress Report

Week 3 October 15, 1987

Automated Regolith Movement System

p. The project was broken down into two main parts: Blower Design and Vehicl e design. Time constraints for the operation were set.

1. Kates, Andy Spoke with Jeff Brown of the University of Houston to find out more about the modules. This information led to finalization of time constraints of the project.

2. Shaw, Chris- Worked on drawings of preliminary blower design. Worked on impact force calculations for projected regolith.

3. Stevens, Anthony- Continued research on patents. Learning the Apollo solid modeling system.

4. Hagler, Ben- Worked on blower design. In the process of learning Apollo solid modeling system. Researched power systems.

5. McWhorter, Valerie- Made graphic of trajectory calculations. Investigating lunar vehicle designs.

6. Dimarco, Mike- Researched lunar surface conditions. Investigating vehicle designs.

7. Porter, Tim- Made graphic of lunar trajectory calculations. Investigated vehicle designs.

8. Crawford, Paul- Investigating power systems.

Progress Report

Week 4 October 22, 1987

Automated Regolith Movement System

The project was broken down into two main parts: Blower Design and Vehicl e design. The design was broken down further with the blower operation seperated into a pickup system and a throwing system, and the vehicle seperated into a power system and a support system.

1. Kates, Andy- Worked on pickup system designs. Investigating power systems.

2. Shaw, Chris- Contacting NASA officials in Huntsville to learn about impact resistance of the modules. Investigating throwing system designs.

3. Stevens, Anthony- Investigating pickup systems for the device. Made some conceptual designs of the whole system and drew them on Autocad.

4. Hagler, Ben- Discussed blower system with a solid handling pump expert. Drew up some prototype configurations.

5. McWhorter, Valerie- Worked on vehicle design including research on treaded vehicles. Made conceptual drawings of system.

6. Dimarco, Mike- Researching lunar surface conditions. Investigating vehicle designs. Researching landscape configurations with respect to vehicle requirments. Working on treaded vehicle design.

7. Porter, Tim- Worked on vehicle design.

8. Crawford, Paul- Discussed pickup systems with respect to soil mechanics with an agricultural engineer.

Progress Report

Week 5 October 29, 1987

Automated Regolith Movement System

Numerical evaluations with regards to power and efficiency for each component of the design were performed. New schemes of regolith collection were investigated.

1. Kates, Andy- Investigated different patterns for the device to follow to gather the required amounts of regolith. Initiated searches for information on lunar soil mechanics and power systems.

2. Shaw, Chris- Made calculations on impellor size, power, and efficiency.

3. Stevens, Anthony- Worked on the design of a disk collection system. Researching apropriate materials for the device.

4. Hagler, Ben- Calculated an approximate impact resistance for the module after discussing such calculations with Dr. McDowell. Worked on the disk collection device.

5. McWhorter, Valerie- Worked on vehicle design including research on treaded vehicles.

6. Dimarco, Mike- Made calculations on power requirements and capabilities of a treaded vehicle.

7. Porter, Tim- Worked on drawings of module area. Worked on vehicle design.

8. Crawford, Paul- Worked on the design of a disk collection device.

Progress Report

Week ~~6~~ 6 ~~October 29~~ November 5, 1987

Automated Regolith Movement System

Progress of each phase of the design was evaluated and a presentation of the project was put together.

1. Kates, Andy- Worked on pickup system designs. Investigating power systems.

2. Shaw, Chris- Contacting NASA officials in Huntsville to learn about impact resistance of the modules. Investigating throwing system designs.

3. Stevens, Anthony- Investigating pickup systems for the device. Made some conceptual designs of the whole system and drew them on Autocad.

4. Hagler, Ben- Discussed blower system with a solid handling pump expert. Drew up some prototype configurations.

5. McWhorter, Valerie- Worked on vehicle design including research on treaded vehicles. Made conceptual drawings of system.

6. Dimarco, Mike- Outlined design requirements of each component of the vehicle and listed possible configurations for each.

7. Porter, Tim- Worked on drawings of module area. Worked on vehicle design.

8. Crawford, Paul- Designed and drew a tiller mechanism for the device.

Progress Report

Week 7 November 12, 1987

Automated Regolith Movement System

The overall configuration of the device is being finalized. Component designs are being drawn up and tested.

1. Kates, Andy- Discussed modeling the disk collection system with Butch Cabe and obtained materials to model the disks with. Worked on disk configuration.

2. Shaw, Chris- Investigating heat transfer and dissipation for the device.

3. Stevens, Anthony- Researching material selection for the various components of the device. Finishing the thrower design.

4. Hagler, Ben- Worked on power system design.

5. McWhorter, Valerie- Working on vehicle design including the bin to hold the regolith.

6. Dimarco, Mike- Working on final design of the vehicle and investigating flexible materials to be used as tracks.

7. Porter, Tim- Worked on vehicle design.

8. Crawford, Paul- Worked on the design of a disk collection device including torque requirements.

Progress Report

Week 8 November 19, 1987

Automated Regolith Movement System

The overall configuration of the device is being finalized. Component designs are being drawn up and tested.

1. Kates, Andy- Built a prototype of soil collection disks and began testing.

2. Shaw, Chris- Researched heat transfer and thermal aspects of the design. Worked on spreadsheet of thrower parameters.

3. Stevens, Anthony- Working on final thrower design. Worked on spreadsheet of pump parameters.

4. Hagler, Ben- Worked on power system design and configurations.

5. McWhorter, Valerie- Working on vehicle design including the bin to hold the regolith.

6. Dimarco, Mike- Working on final design of the vehicle and investigating flexible materials to be used as tracks.

7. Porter, Tim- Worked on bin design.

8. Crawford, Paul- Worked on the design of a disk collection device including torque requirements.

Appendix 10

Disclosure of Invention

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Record of Invention No.__________
UTC No. (if applicable)__________

GEORGIA INSTITUTE OF TECHNOLOGY

APPROVAL SHEET (Attach to DISCLOSURE OF INVENTION)

The following questions should be answered by the laboratory or school director, as applicable. The questions are designed to verify the ownership of the invention. This approval should be included when the Invention Disclosure form is submitted to the Office of Technology Transfer.

1. Title of Invention:

AUTOMATED REGOLITH MOVEMENT SYSTEM (A.R.M.S.)

2. List of Inventor(s):

PAUL CRAWFORD	VALERIE McWHORTER
MICHAEL DiMARCO	TIMOTHY PORTER
BEN HAGLER	CHRISTOPHER SHAW
ANDREW KATES	ANTHONY STEVENS

3. Ownership:

In my opinion this invention:

__X__ A. Is owned by the Institute in accordance with the Patent Policy.

_____ B. Was developed by the inventor(s) without use of Institute time, facilities or materials, and is not related to the inventor's area of technical responsibility to the Institute and hence belongs to the inventor(s).

4. Research project advisor approval for student submissions (if applicable):

______________________________ Advisor ______________ Date

Reviewed for Institute ownership by laboratory or school director.

______________________________ Name ______________ Date

______________________________ Title/Unit

Record of Invention No.______ 147
UTC No. (if applicable)______

GEORGIA INSTITUTE OF TECHNOLOGY
DISCLOSURE OF INVENTION

Submit this disclosure to the Office of Technology Transfer (OTT) or contact that office for assistance. Disclosure must contain the following items: (1) title of invention, (2) a complete statement of invention and suggested scope, (3) results demonstrating that the concept is valid, (4) variations and alternate forms of the invention, (5) a statement of the novel features of the invention and how these features distinguish your invention from the state of the art as known to you, (6) applications of the technology, and (7) supporting information.

1. Title

Technical Title: AUTOMATED REGOLITH MOVEMENT SYSTEM

Layman's Title (34 characters maximum, including spaces): SELF-RUN LUNAR SOIL THROWER

Inventor(s): (Correspondence, patent questions, etc. will be directed to the first named inventor)

A. Signature Andrew R. Kates Revenue Share% 33 Date 12-2-87

Printed Name ANDREW RICHARD KATES Citizenship U.S.
(First Middle Last)

Home Address 15 BIRCH RISE

City NEWTOWN County FAIRFIELD State CT Zip Code 06470

Campus Unit/Mail Address GA. TECH P.O. Box 34520 Campus Phone (404) 676-1543

B. Signature Benjamin L. Hagler Revenue Share% 33 1/3 Date 12-2-87

Printed Name BENJAMIN LENOIR HAGLER Citizenship U.S.
(First Middle Last)

Home Address ROUTE 2 BOX 690

City THOMSON County MCDUFFIE State GA Zip Code 30824

Campus Unit/Mail Address GA. TECH P.O. Box 31437 Campus Phone (404) 325-3126

C. Signature Anthony J. Stevens Revenue Share% 33 1/3 Date 12-2-87

Printed Name ANTHONY JEROME STEVENS Citizenship U.S.
(First Middle Last)

Home Address 2600 WOLFE STREET

City BRUNSWICK County GLYNN State GA Zip Code 31520

Campus Unit/Mail Address GA. TECH P.O. Box 37267 Campus Phone (404) 886-7294

Record of Invention No._____
UTC No. (if applicable)_____

DISCLOSURE OF INVENTION

2. Statement of Invention:

Give a complete description of the invention. If necessary, use additional pages, drawings, diagrams, etc. Description may be by reference to a separate document (copy of a report, a preprint, grant application, or the like) attached hereto. If so, identify the document positively. The description should include the best mode that you presently contemplate for making (the apparatus or material invented) or for carrying out the process invented.

REFER TO DESIGN REPORT ATTACHED HERETO.

Inventor(s) ANDREW R. KATES Date 12/2/87

BENJAMIN L. HAGLER Date 12/2/87

ANTHONY J. STEVENS Date 12/2/87

Witness* __________ Date ______

__________ Date ______

*The witness should be technically competent and understand the invention.

DISCLOSURE OF INVENTION

3. Results Demonstrating the Concept is Valid:

Cite specific results to date. Indicate whether you have completed preliminary research, laboratory model, or prototype testing.

SORTING DISCS EFFICIENCY TEST

- TEST WERE RUN TO DETERMINE THE SORTING CAPABILITIES OF THE SORTING DISCS. TESTS WERE PERFORMED IN RELATIVELY LOOSE EARTH SOIL TO SIMULATE THE LUNAR SOIL. RESULTS CONFIRMED THAT THE DISCS ~~WERE~~ SHOULD BE ADEQUATE FOR SORTING THE LUNAR SOIL.

4. Variations and Alternative Forms of the Invention:

State all of the alternate forms envisioned to be within the full scope of the invention. List all potential applications and forms of the invention, whether currently proven or not. (For example, chemical inventions should consider all derivatives, analogues, etc.) Be speculative in answering this section. Indicate what testing, if any, you have conducted on these alternate forms.

THE INVENTION MAY BE USED FOR MOON SITE PREPARATION. AND THE TRANSPORTATION OF MATERIALS AND EQUIPMENT ON THE MOON

Inventor(s) Andrew Kates Date 12/2/87

Benjamin Lenoir Haylor Date 12/2/87

Anthony J. Stevens Date 12/2/87

Witness* ______________________ Date ________
(printed name)

______________________ Date ________
(printed name)

DISCLOSURE OF INVENTION

5. Novel Features:

a. Specify the novel features of your invention. How does the invention differ from present technology?

THE SORTING DISCS REPRESENTS A NEW METHOD FOR SORTING MATERIAL. THE DISCS ARE A VERY SIMPLE WAY OF SORTING MATERIALS

b. What deficiencies or limitations in the present technology does your invention overcome?

THE DISCS OVERCOME THE NEED FOR HEAVY MACHINERY TO SORT MATERIALS

c. Have you or an associate searched the scientific literature with respect to this invention? Yes X No______. Have you done a patent search? Yes X No______. If yes in either case, or both, indicate what pertinent information you found and enclose copies if available. Also indicate any other art you are aware of (whatever the source of your information) that is pertinent to your invention. Enclose copies of descriptions if available. (Note: An inventor is under duty by law to disclose to the U.S. Patent and Trademark Office any prior art known to him or her.)

INFORMATION WAS SEARCHED ON THE VEHICLE DRIVE SYSTEM AND THE PUMPING MECHANISM. FOR THE VEHICLE DRIVE SYSTEM, ARMORED VEHICLE DESIGN WERE RESEARCHED AND SOLID HANDLING PUMPS FOR THE PUMPING MECHANISM.

Inventor(s) Andrew Kates Date 12/2/87

Benjamin Lorain Haylor Date 12/2/87

Anthony J. Stevens Date 12/2/87

Witness ______________________ Date______
(printed name)

______________________ Date______
(printed name)

Record of Invention No.________
UTC No. (if applicable)________

DISCLOSURE OF INVENTION

6. Application of the Technology:

List all products you envision resulting from this invention. For each, indicate whether the product could be developed in the near term (less than 2 years) or would require long-term development (more than 2 years).

NONE

Inventor(s) Andrew Kate Date 12/2/87
Benjamin Feraie Hopfor Date 12/2/87
Anthony J. Stevens Date 12/2/87

Witness ________________________ Date________
(printed name)
________________________ Date________
(printed name)

Record of Invention No.________
UTC No. (if applicable)________

DISCLOSURE OF INVENTION
SUPPORTING INFORMATION

1. Are there publications such as theses, reports, preprints, reprints, etc. pertaining to the invention? Please list with publication dates. Include manuscripts (submitted or not), news releases, feature articles and items from internal publications. Supply copies if possible.

2. On what date was the invention first conceived? 10/29/87 Is this date documented? YES Where? WEEKLY PROGRESS REPORT Are laboratory records and data available? Give reference numbers and physical location, but do not enclose.

GEORGIA TECH, MR. JIM BRAZELL, INSTRUCTOR OF MECH. ENG.

3. Give date, place, and circumstances of any disclosure. If disclosed to specific individuals, give names and dates.

4. Was the work that led to the invention sponsored by an entity external to Georgia Institute of Technology? Yes X No________

a) If yes, has sponsor been notified? Yes X No________

b) Sponsor Names: NASA

GIT Project Nos.

5. What firms do you think may be interested, in the invention and why. Name specific persons within the companies if possible.

Record of Invention No._______
UTC No. (if applicable)_______

DISCLOSURE OF INVENTION
SUPPORTING INFORMATION

6. Setting aside your personal interest, what do you see as the greatest obstacles to the adoption of your invention?

   AN EFFECTIVE TEST ON THE MOON

7. Alternate Technology and Competition:

   a. Describe alternate technologies of which you are aware that accomplish the purpose of the invention.

   b. List the companies and their products currently on the market which make use of these alternate technologies.

   c. List any research groups currently engaged in research and development in this area.

      NASA

8. Future Research Plans:

   a. What additional research is needed to complete development and testing of the invention? What time frame and estimated budget is needed for the completion of each step?

   b. Is this additional research presently being undertaken? Yes___ No X

   c. If yes, under whose sponsorship?

   d. If no, should corporate sponsorship be pursued? Yes \\ No X

      Suggested corporation(s)________________

9. Attach, sign and date additional sheets if necessary. Enclose sketches, drawings, photographs and other materials that help illustrate the description. (Rough artwork, flow sheets, Polaroid photographs and penciled graphs are satisfactory as long as they tell a clear and understandable story.)